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Vol. 197: Journal Section of the American Society of Heating and Ventilating Engineers

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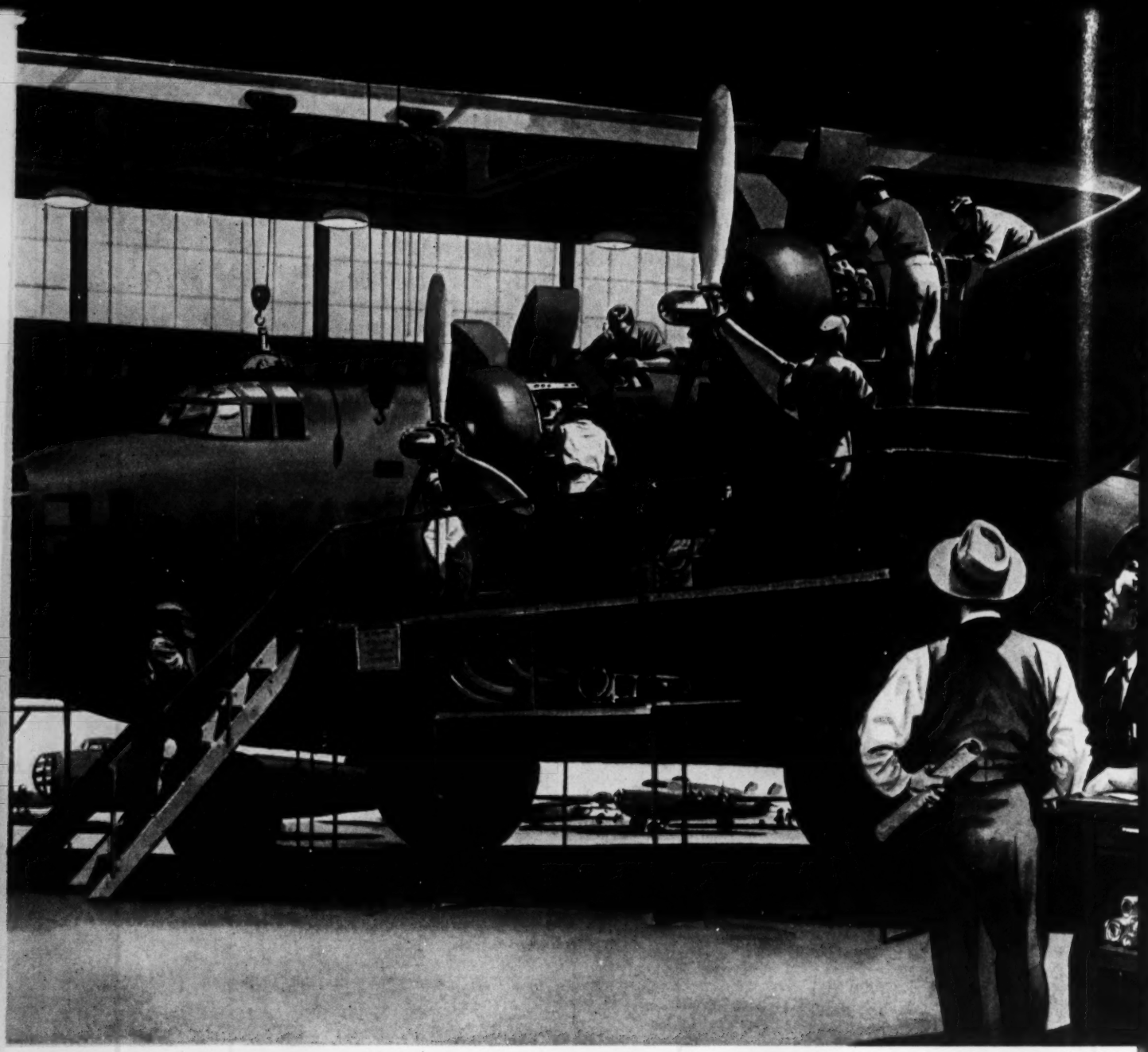

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The Editor's Pages

SUPPLEMENTAL OIL FOR "HARDSHIP" CASES

Those who face serious hardship under fuel oil rationing due to abnormal conditions beyond their control may apply at their local boards for supplemental allotments, the Office of Price Administration announced last month. This supplemental ration procedure covers all residential buildings, including private homes, apartment houses, and cold water flats. The new procedure provides that local boards may recommend to the appropriate OPA district office the issuance of additional coupons for residential dwellings when the consumer meets certain specified conditions and shows that the remaining ration is insufficient to maintain minimum heating needs.

At the same time, OPA announced that private homeowners, residents of multiple dwellings heated by oil stoves, and all commercial users may apply for an adjustment of their present ration to offset changes in circumstances which increase their heating or hot water needs in certain limited instances. The first arrangement is designed to augment the rations of residential structures where, although basic heating needs have not increased, the original ration is insufficient due to abnormally cold weather this winter or to like developments.

The second will increase the ration of householders (with exception of those living in apartment houses heated by central plants) and commercial users where the original allotment has proved too small due to an increase in basic heating needs. Such instances may include cases where there is an increase in the number of people living in a private home or where a commercial establishment is using more heating oil due to extension in hours of operation.

Banks, factories, office buildings, and all other commercial users of oil for heating may apply for an adjusted ration if their heat or hot water requirements have increased because of longer hours of operation or similar change of circumstances. In addition, they must show that they have taken all possible steps to reduce their consumption and that they cannot convert their equipment or use coal or wood stoves.

The ration mechanism provides that consumers eligible for adjustment may apply at their local board for a substitute ration any time during the present heating year—that is, until September 30. A new application is to be filled out using the same form as used when applying originally (OPA R-1100 or 1101).

This action is contained in amendments Nos. 45 and 46 to ration order 11, effective March 13.

MAXIMUM PRICE ORDER 251 TO BE REVOKED

Most parts of the construction industry not already exempt from price control soon will be removed from regulation, Price Administrator Prentiss M. Brown announced March 22. However, repair and maintenance services which are a factor in civilian rents will remain under control.

All general contracting activity and many types of subcontracting for jobs on new dwellings and factories as well as many of the services connected with the construction of a new building will be formally excluded from price regulation. The administrator stated that the removal from price control would be accomplished through revocation of maximum price regulation No. 251 (construction services and sales of building and industrial equipment and materials on an installed or erected basis).

The necessity for a specific regulation over the entire construction field has been eliminated almost completely by the influence of wartime limitations on civilian construction and the drastic decline of contracting activity, Mr. Brown stated. However, certain construction activities which are factors in the preservation of rent ceilings will remain under control of other price measures.

The administrator pointed out that most of the industry is now active in military construction and other government building activities which previously have been exempted from regulation. The remaining construction activities, which were subject to OPA control, will be exempted, except in cases where transfer to other regulations will be announced. Examples of construction work which will remain under control are as follows:

Repair services, including repair and maintenance of heating plants.

Sheet metal work.

The sale of goods—such as roofing, flooring, and siding—on an installed basis when such materials are necessary for the maintenance of existing structures.

The sale of building materials on an installed basis where the items are sold on a unit basis. Materials such as pipe and fencing, and mechanical equipment such as water heaters and furnaces, when sold on an installed unit basis, will remain under control.

The prices for materials purchased by the industry will also remain subject to applicable price regulations and wages paid—subject to applicable wage controls of the National War Labor Board and the Wage Adjustment Board.

ORDER L-42 ON PIPE FLANGES AMENDED

To reduce the demand for steel flanges, schedule 2 of limitation order L-42 has been amended to permit greater use of grey cast iron and malleable iron in pipe flange manufacture. WPB officials pointed out that order L-252 permits malleable iron bodies for several pressure classes of valve. The malleable iron screwed and lapped joint flanges listed in schedule 2 will become available for the malleable iron body valves permitted under L-252.

The amended schedule lists the sizes of pipe flanges, in the production of which grey cast iron or malleable iron are allowed. It also permits the production of malleable iron screwed flanges, blind flanges, reducing screwed flanges, and lap joint flanges in those pipe sizes in which grey cast iron is allowed. Grey cast iron lap joint flanges are permitted in additional pipe sizes.

C. E. SCOTT HEADS UNIT HEATER ASSOCIATION

At the seventeenth annual meeting of the Industrial Unit Heater Association, held in Detroit last month, C. E. Scott was elected president, E. W. Petersen was elected vice-president, and L. O. Monroe was named secretary-treasurer.

FIREPLACE CALLED AN "EFFICIENT EXHAUST FAN"

E. P. Heckel, of E. P. Heckel and Associates, a member of HPAC's board of consulting & contributing editors, has been active in fuel oil rationing work as a consultant to a local board. He noticed that many oil users who had fireplaces in their homes were running over their oil rations, and decided to do some investigating.

Results of an informal study of one house, for which complete plans were available and in which the fireplace was in use constantly, indicated that as much as 700 to 800 cfm of air was drawn into the fireplace, heated, and wasted up the chimney. Mr. Heckel concludes that the ordinary fireplace is a fairly efficient exhaust fan, and that its use increases rather than reduces oil consumption for heating the building.

The Editor's Pages

NEW ENGLAND OIL SALES 76% OF NORMAL

Heating oil sales in New England by the 13 principal supplying companies amounted to 75.9 per cent of normal during the four month period, October through January, it was announced last month by Petroleum Administrator for War Harold L. Ickes.

Normal demand was calculated on a basis of a weighted average of the temperatures in four principal New England cities — Boston, Portland, Hartford and Providence.

CHANGE TO HAND FIRING JUMPS COSTS

In anticipation of the threatened and now very much evident oil shortage, boiler plants in 13 buildings of the New York Telephone Co. in Manhattan were converted to coal burning last summer, as a result of which fuel oil requirements for heating were reduced by over 90 per cent.

Boilers were converted to hand firing anthracite coal burning. Consequently, it was necessary to employ firemen during the heating season and, in view of the full time nature of telephone service, constant boiler attendance is required at the larger buildings to maintain around-the-clock heating. Based on experience during the current heating season, the expense of operating the heating boilers with coal is about 50 per cent greater than that with oil, the increase being chiefly attributable to additional labor requirements, according to George N. Butz, superintendent of buildings and supplies.

WAR ACTIVITIES OF CONTROL MAKER REPORTED

The Minneapolis-Honeywell Regulator Co., although totally without previous experience in the field, has in the space of two years become one of the country's leading volume producers of ordnance fire control equipment, Harold W. Sweatt, president, announced in the annual report mailed to stockholders last month.

The company's standard line products were made during the year in restricted quantities to fill the needs for heating, air conditioning, and other controls for Army camps, war plants, government buildings, and other priority purposes, as well as for permissible service and maintenance of existing installations, the report said. The Brown Instrument Co. division operated at a high level throughout the year to meet a constantly growing demand for industrial control instruments for war production.

In connection with its production of heating control equipment, the company volunteered to conduct schools for Army personnel in the design, operation, and servicing of heating controls and systems, whether manufactured by it or others, to make possible the most efficient operation of the Army's installations. During 1942 more than 10,000 uniformed men went through schools conducted at each major camp in the country, it was stated.

FACES POST WAR PERIOD WITH CONFIDENCE

Controverting those who say that America must have an inevitable collapse after the war, or that government must take over the job of providing the things that people need and want, Cloud Wampler, president of Carrier Corp., last month asserted his belief that American industry can provide good standards of living in the post war period. Mr. Wampler spoke on a national radio network.

"Every war has brought advances in technology—new products—new horizons. The present conflict transcends all others in this respect. When the war is over, a multitude of scientific developments will be harnessed by industry and there will be generated an advanced type of industrial revolution," he said.

"During the war there is being created the greatest pent-up demand for goods—both consumer and durable—that this country or any other has ever seen," he pointed out.

Carrier Corp., a leading manufacturer of air conditioning and refrigeration equipment, recently reported for the fiscal year ended October 31, 1942, net income of $541,564, equivalent after all charges including taxes to $1.33 per share. The report states that 95 per cent of the business written by the corporation during fiscal 1942 was for war purposes. In this work, standard products are normally used, although the requirements peculiar to numerous jobs have led to many special designs.

DRAFT DELAYS FOR ENGINEERING STUDENTS

According to a directive issued by the national selective service headquarters last month, freshmen college students in engineering are to be allowed draft deferments until graduation. The directive covers students who will complete their work before July 1, 1945.

Among the courses of study included are heating, ventilating, refrigerating, and air conditioning engineering.

VENTILATION FOR SHIPS UNDER CONSTRUCTION

A slight error appeared in the drawing on p. 711 of the article Ventilation for Ships Under Construction, in the December, 1942, HPAC. Diameters of the adapter for the 7000 cfm fan were given as 9 in. and 16 in., which are incorrect as the fan exhaust port is 22 in. The 9 and 16 in. dimensions are correct for 2500 cfm fans, which size is used wherever possible.

GOOD NEWS FOR FUEL OIL DEALERS

Envelopes will replace the gummed sheets, form OPA R-120, for fuel oil coupons when current supplies of the sheets are exhausted. General ration order No. 7 allows for the use of sealed envelopes and specifies the manner in which they must be used. Only coupons of the same type and value and with the same validity period may be enclosed in the same envelope.

The person who encloses the coupons in the envelope must inscribe in ink on the face of the envelope the number and type of coupons enclosed, their total value and the represented commodity (fuel oil), together with his name (or company) and address. In addition, he must sign his name across the sealing flap of the envelope, indicating certification of the statements written on the face.

1860 MALCOLM McDOWELL 1943

By V. L. Sherman

If air conditioning and other scientific research ever had "a friend at court" that one was Malcolm McDowell, one of the finest and best loved of all newspaper men. His ability to translate technical and complicated phraseology into terms an ordinary mortal could understand was most unusual, and saved from obscurity many commendable projects.

Mr. McDowell covered the World's Columbian Exposition in Chicago in 1893 for the Chicago Record. In 1933 and 1934, at the age of 72, Mr. McDowell covered the Century of Progress in Chicago for the Chicago Daily News. He knew of the progress in industrial air conditioning and, with his always ready background, publicized residential air conditioning in a way the public could understand. No man in the industry ever did more to promote its healthy growth.

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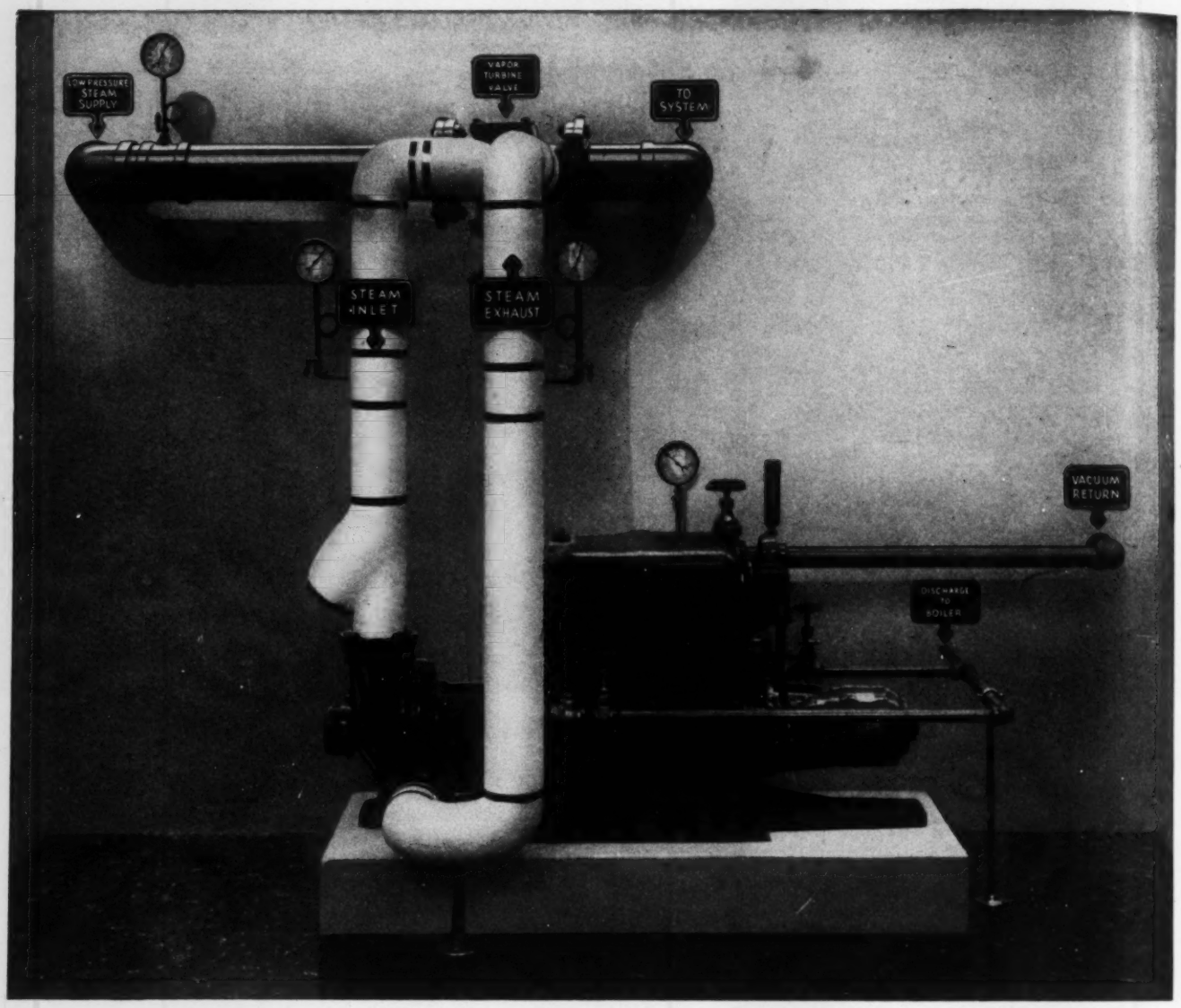
Interrupted Electric Service will not stop this NASH Heating Pump!



In Hospitals, Greenhouses, Schools, Public Buildings, Theatres, wherever heating systems must not fail, install the Nash Vapor Turbine, for it is independent of electric current failure, and continues to operate as long as there is steam in the system.

This is because the prime motive power of this economical pump is a special steam turbine, controlled by a unique "Vapor Turbine Valve", which automatically bypasses from the heating main a small portion of steam, the exact amount necessary to develop the power needed to remove the condensate and maintain the required vacuum on the system. Even this small amount is passed immediately back to the mains, and goes on to the system with little heat loss. This pump operates on any system, high or low pressure.

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THE NASH ENGINEERING COMPANY

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WATER SPRAYED DIRECT EXPANSION COILS

• Their Theory and Characteristics •

Although Used Extensively in the Manufacture and the Testing of War Goods, the Fundamentals of Such Coils Have Never Before Been Analyzed as Completely as in This Treatment by William Goodman

SUMMARY—Sprayed direct expansion coils are widely used in air conditioning and refrigeration work. Coils sprayed with water are used in air conditioning, and brine sprayed coils are, of course, used in low temperature air conditioning and refrigeration installations. Because such coils are used extensively today for all kinds of manufacturing and testing in connection with the war effort, a fundamental treatment of their theory and characteristics is especially timely. In spite of the fact that such coils have been used for many years, such a treatment has never before been published. . . . In the article of which this is the first part, the theory and characteristics of water sprayed coils are covered in a simple and thorough fashion with many illustrative examples to show not only how the characteristics of such coils are computed, but also to show the effect of variations in the various quantities which affect the performance. A thorough understanding of the basic characteristics and theory of sprayed coils should result in much more efficient and satisfactory installations. . . . Mr. Goodman is consulting engineer, the Trane Co., and a member of HPAC's board of consulting & contributing editors

DIRECT EXPANSION coils sprayed with either water or brine are extensively used in air conditioning and refrigeration. Water sprayed coils are frequently used for above-freezing-temperature installations, especially where cleanliness and freedom from odors are important. Brine sprayed coils, of course, are used for low temperature refrigeration and air conditioning installations.

When coils are sprayed, the water or brine is recirculated continuously from the tank below the coil to the spray nozzles as shown in Fig. 1. The water is neither heated nor cooled as it flows through the piping between the tank and the nozzles. The heat added by the pump itself is negligible compared to all the other quantities involved and may be ignored for the purposes of the present discussion. Although vertical air flow through a horizontal coil is shown in Fig. 1, equipment in which the air flows horizontally through vertical coils is quite common.

The air as it flows through the sprayed coil surrenders heat and moisture to the water film covering the coil. But the water, as fast as it receives this heat, transfers it in turn to the boiling refrigerant inside the tubes. Because the water is neither heated nor cooled as it flows through the piping, it is obvious that the water must attain an equilibrium temperature which remains constant for any one given set of operating conditions.

Because the water temperature remains constant, the change in the condition of the air as it flows through a sprayed coil can be represented by a straight line on a psychrometric chart[1] whose coordinates are specific humidity and enthalpy (total heat).

Whenever water and air are in direct contact with each other, the change in the condition of the air can be represented by a straight line on the psychrometric chart if the temperature of the water remains constant[2]. Hence, in the sprayed cooling coil, because the temperature of the water is constant, the change in the condition of the air as it flows through the coil can be represented by a straight line. As is shown diagrammatically on the skeleton psychrometric chart of Fig. 2, the straight line is drawn between the point on the chart representing the initial condition of the air and the point on the saturation curve intersected by the dry bulb temperature line which is equal to the constant temperature of the spray water. In Fig. 2, point *1* represents the initial condition of the

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[1]For such a chart, see HPAC, June, 1939, p. 359. A chart of this kind is also included in the book Air Conditioning Analysis, published by MacMillan. In this article all further references to psychrometric charts will be understood to refer to these charts. Furthermore, values of enthalpy (total heat) used in solving examples throughout this article are obtained from the tables in the above mentioned book. The values of total heat in the book are always 10 Btu higher than the values in the psychrometric tables published in HPAC, January to April, 1939. Ten Btu was added to all values in order to eliminate negative values of total heat at low temperatures. Incidentally, this same difference of 10 Btu will be observed for corresponding points on the psychrometric charts in HPAC and the psychrometric charts in the book.

[2]p. 173, Air Conditioning Analysis. Also HPAC, May, 1938, p. 328.

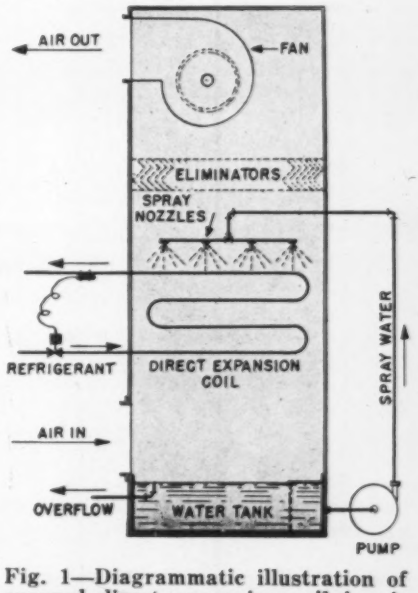


Fig. 1—Diagrammatic illustration of sprayed direct expansion coil in air conditioning unit. Although this illustration shows a horizontal coil with vertical air flow, vertical coils with horizontal air flow are also used

air entering the coil, point *3* represents the constant temperature of the spray water, and point *2* represents the final condition of the air leaving the coil. Point *2*, representing the condition of the leaving air, can lie anywhere between points *1* and *2*. Its exact location depends upon the coefficients of heat transmission, the amount of surface, and the amount of air. The influence of these quantities upon the location of point *2*, and hence upon the capacity of the sprayed coil, will be discussed in detail, and methods will be presented for calculating these quantities as well as the location of point *2* on Fig. 2. For the present, the discussion will be confined entirely to water.

A line like *1-3* of Fig. 2 will for convenience be called a *condition line*. For a given spray water temperature and initial condition of the air, all of the points representing the possible final conditions of the air leaving sprayed coils of various areas must lie on the condition line.

Total Heat Removed by Coil

Although heat is removed from the air and moisture condensed from it, there is no need to evaluate separately the transfer of heat and moisture from the air to the water. Instead, the total heat transferred—which includes both the sensible heat surrendered by the air and the latent heat of the moisture—depends only upon the wet bulb temperature of the air and the temperature of the spray water. The heat transferred from the air to the water can be computed by means of the following equation:

$$Q = G\ (h_1 - h_w)\ Z \quad \ldots\ldots\ldots [1]$$

This equation is derived in the May, 1938, issue of HPAC, p. 327, appendix 1.

The value of the factor Z can be found in Table 1 after first computing another factor, M, by means of the following formula:

$$M = \frac{f_s}{0.24}\,\frac{A}{G} \quad \ldots\ldots\ldots\ldots [2]$$

The factor Z might be called the "efficiency" of the coil because it rep-

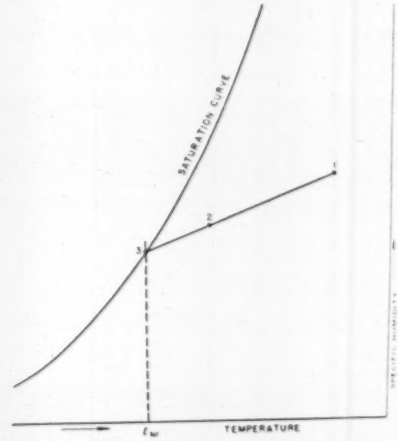


Fig. 2—The cooling of air in a sprayed coil takes place along a straight line on the psychrometric chart. Point *1* represents the initial condition of the air entering the coil; point *2*, the final condition of the air leaving the coil; and point *3*, the temperature of the spray water. Point *2*, representing the final condition of the air, can lie anywhere along line *1-3*, depending upon the area of the coil. Line *1-3* is called the *condition line* because the points along it represent the final condition of the air

Table 1 — Values of Z and N_L/N for Values of M

M	Z	N_L/N	M	Z	N_L/N	M	Z	N_L/N
0.00	0.0000	1.0000	0.45	0.3624	0.8053	1.30	0.7275	0.5596
.01	.0100	0.9950	.46	.3687	.8016	.32	.7329	.5552
.02	.0198	0.9900	.47	.3750	.7979	.34	.7381	.5509
.03	.0296	0.9851	.48	.3812	.7942	.36	.7433	.5466
.04	.0392	0.9803	.49	.3874	.7906	.38	.7484	.5423
0.05	0.0488	0.9754	0.50	0.3935	0.7869	1.40	0.7534	0.5381
.06	.0582	.9706	.52	.4055	.7798	.42	.7583	.5340
.07	.0676	.9658	.54	.4173	.7727	.44	.7631	.5299
.08	.0769	.9610	.56	.4288	.7657	.46	.7678	.5259
.09	.0861	.9563	.58	.4401	.7588	.48	.7724	.5219
0.10	0.0952	0.9516	0.60	0.4512	0.7520	1.50	0.7769	0.5179
.11	.1042	.9470	.62	.4621	.7453	.55	.7878	.5082
.12	.1131	.9423	.64	.4727	.7386	.60	.7981	.4988
.13	.1219	.9377	.66	.4831	.7320	.65	.8080	.4897
.14	.1306	.9332	.68	.4934	.7256	.70	.8173	.4808
0.15	0.1393	0.9286	0.70	0.5034	0.7192	1.75	0.8262	0.4721
.16	.1479	.9241	.72	.5132	.7128	.80	.8347	.4637
.17	.1563	.9196	.74	.5229	.7066	.85	.8428	.4555
.18	.1647	.9152	.76	.5323	.7004	.90	.8504	.4476
.19	.1730	.9107	.78	.5416	.6944	.95	.8577	.4399
0.20	0.1813	0.9063	0.80	0.5507	0.6883	2.00	0.8647	0.4323
.21	.1894	.9020	.82	.5596	.6824	.10	.8775	.4179
.22	.1975	.8976	.84	.5683	.6765	.20	.8892	.4042
.23	.2055	.8933	.86	.5768	.6707	.30	.8997	.3912
.24	.2134	.8890	.88	.5852	.6650	.40	.9093	.3789
0.25	0.2212	0.8848	0.90	0.5934	0.6594	2.50	0.9179	0.3672
.26	.2289	.8806	.92	.6015	.6538	.60	.9257	.3560
.27	.2366	.8764	.94	.6094	.6483	.70	.9328	.3455
.28	.2442	.8722	.96	.6171	.6428	.80	.9392	.3354
.29	.2517	.8681	.98	.6247	.6374	.90	.9450	.3259
0.30	0.2592	0.8639	1.00	0.6321	0.6321	3.00	0.9502	0.3167
.31	.2666	.8598	.02	.6394	.6269	.10	.9550	.3080
.32	.2739	.8558	.04	.6465	.6217	.20	.9592	.2998
.33	.2811	.8517	.06	.6535	.6166	.30	.9631	.2919
.34	.2882	.8477	.08	.6604	.6115	.40	.9666	.2843
0.35	0.2953	0.8437	1.10	0.6671	0.6065	3.60	0.9727	0.2702
.36	.3023	.8398	.12	.6737	.6015	.80	0.9776	.2573
.37	.3093	.8359	.14	.6802	.5966	Inf.	1.0000	.2512
.38	.3161	.8319	.16	.6865	.5918			
.39	.3229	.8281	.18	.6927	.5871			
0.40	0.3297	0.8242	1.20	0.6988	0.5823			
.41	.3364	.8204	.22	.7048	.5777			
.42	.3430	.8166	.24	.7106	.5731			
.43	.3495	.8128	.26	.7163	.5685			
.44	.3560	.8090	.28	.7220	.5640			

Symbols

A = area, in sq ft;

f_s = coefficient of heat transfer between air and water (through air film), Btu per hr per deg per sq ft;

f_R = coefficient of heat transfer between wall of metal tube and evaporating refrigerant (refrigerant film coefficient), Btu per hr per deg per sq ft;

f_w = coefficient of heat transfer between water and metal wall of tube (water film coefficient), Btu per hr per deg per sq ft;

G = weight of dry air through coil, lb per hr;

G_o = weight of outdoor air supplied to conditioned space, lb per hr;

h_1 = initial enthalpy (total heat content) of air entering sprayed coil, Btu per lb of dry air;

h_w = enthalpy (total heat content) of saturated air at a temperature equal to the temperature of the spray water, Btu per lb of dry air;

M = a quantity defined by equation 2;

N = a quantity defined by equation 6;

N_L = limiting value of N as defined by equation 8;

Q = total quantity of heat removed from the air, Btu per hr;

R = ratio of external to internal surface of coil;

t_R = temperature of refrigerant, F;

t_w = temperature of spray water, F;

t'_1 = initial wet bulb temperature of air, F;

U = overall coefficient of heat transfer from spray water to evaporating refrigerant, Btu per hr per deg per sq ft (defined by equation 4);

Z = a quantity that depends on M (see Table 1);

θ = angle on spray water-air chart (see Fig. 4).

resents the ratio of the actual heat transferred through a coil to the amount that could be transferred through a coil having either an infinite amount of surface or an infinite coefficient of heat transfer between the air and the water.

The amount of heat transferred from the air to the water is, of course, equal to the amount of heat which is transferred in turn from the water to the boiling refrigerant inside the coil. The amount of heat transferred from the water to the refrigerant can be computed by means of the following equation:

$$Q = U\ (t_w - t_R)\ A \quad \ldots\ldots\ldots [3]$$

In this equation the value of U is computed by means of

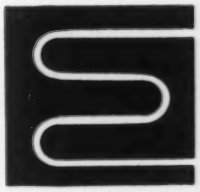
$$\frac{1}{U} = \frac{1}{f_w} + \frac{R}{f_R} \quad \ldots\ldots\ldots\ldots [4]$$

The derivation of equations 3 and 4 is given in HPAC, May, 1938, p. 327, appendix 2.

The total heat removed from the air by the coil can be computed by means of either equation 1 or equation 3. The heat transfer coefficient f_g in equation 2 and the coefficients f_w and f_R must be determined experimentally or estimated from whatever data can be found in the published literature.

Spray Water Temperature

Before either equation 1 or equation 3 can be used, the temperature of the spray water must be known.



As is shown below, for a given coil having a given coefficient of heat transfer and area, and a given quantity of air, the temperature of the spray water depends only upon the initial wet bulb temperature of the air and the constant temperature of the boiling refrigerant. Dividing equation 1 by equation 3, the following equation is obtained.

$$\frac{h_1 - h_w}{t_w - t_R} = \frac{U\ A}{Z\ G} \quad \ldots\ldots\ldots [5]$$

For convenience, let

$$N = \frac{U\ A}{Z\ G} \quad \ldots\ldots\ldots\ldots\ldots [6]$$

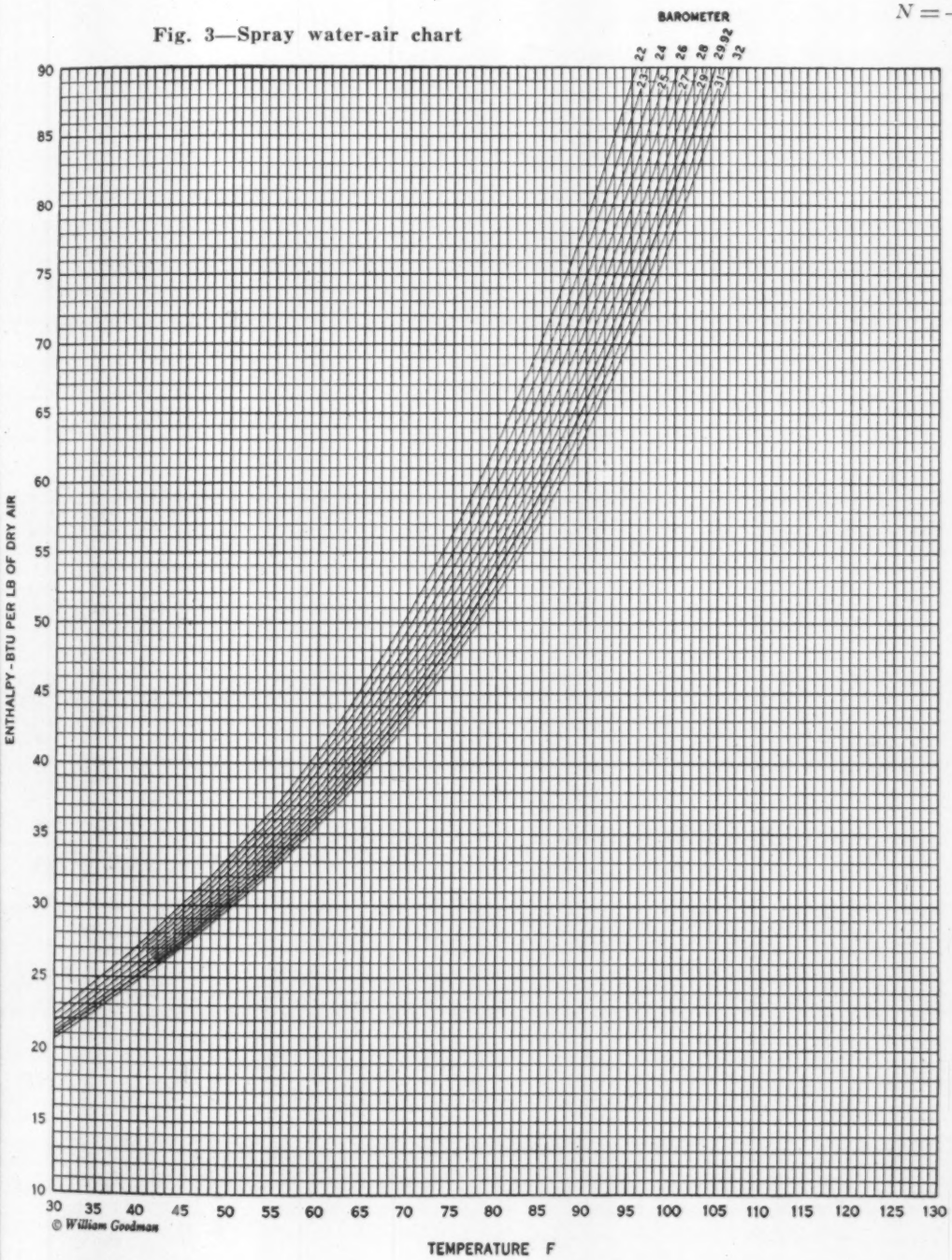
Hence,

$$\frac{h_1 - h_w}{t_w - t_R} = N \quad \ldots [7]$$

In solving equation 7, two cases arise in practice. In the first case, the area of the coil and the air quantity are given as well as the various coefficients of heat transfer. In this case, the temperature of the spray water is to be found for a given wet bulb temperature of the entering air and temperature of the refrigerant. In the second case, the temperature of the spray water is assumed for given entering wet bulb and refrigerant temperatures. The problem is then to find either the coil area or the air quantity corresponding to the given spray water temperature. The method of solving the first case in which the temperature of the spray water is to be found for a given area and air quantity is discussed in the following paragraphs. The solution of the second case will be discussed later.

If the area of the coil, the quantity of air, and the other quantities needed to find the value of N are given, equation 7 can be used to compute the temperature of the spray water corresponding to a given initial wet bulb temperature and refrigerant temperature.

Fig. 3—Spray water-air chart



However, inasmuch as both h_w and t_w appear in equation 5, this equation cannot conveniently be solved by ordinary algebraic methods; trial and error must be used. However, the spray water-air chart of Fig. 3[3] can be used to find the spray water temperature directly.

Equation 7 can be represented by a straight line on the water-air chart[4] of Fig. 3. The slope of this straight line depends only upon the value of N. This is illustrated diagrammatically in Fig. 4, where line *3-4* is a straight line representing equation 7 drawn at an angle Θ which depends only upon the value of N. The actual value of the angle Θ for any value of N can be found in Table 2.

To determine the spray water temperature, first compute the value of N for the coil by means of equation 6. Next find the corresponding value of Θ in Table 2. Then referring to Fig. 4, which illustrates the method of using the water-air chart, locate point *3* at the intersection of the horizontal line representing the initial enthalpy (heat content) of the air and the vertical line representing the constant temperature of the boiling refrigerant. Through point *3* draw a diagonal line at an angle Θ with the horizontal. Point *4* is located where this diagonal line intersects the saturation curve for the given barometric pressure — ignore the curves for all other barometric pressures. The temperature of the spray water is read on the vertical line through point *4*.

The initial total heat content of the air depends very nearly upon the initial wet bulb temperature of the air. Consequently, if the initial wet bulb temperature of the air is given, the horizontal line corresponding to its enthalpy can be found directly on the water-air chart as illustrated in Fig. 4. First locate, on the scale of temperature, point *1* representing the initial wet bulb temperature of the air. Locate point *2* where the vertical line of temperature through point *1* intersects the saturation curve for the given barometric pressure. The horizontal line through point *2* represents the initial enthalpy of the air. This method of using the approximate enthalpy instead of the true enthalpy will be used throughout this article. For more accurate results, the true enthalpy corresponding to the actual initial condition of the air should be used. The true enthalpy can be read either from the psychrometric chart or computed from data in psychrometric tables.[5]

[3]Reproduced from Air Conditioning Analysis.

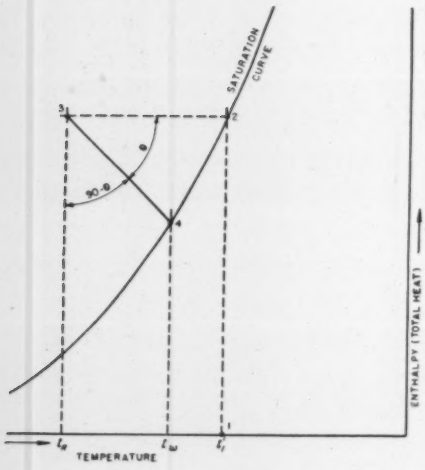
[4]p. 205, Air Conditioning Analysis.

[5]p. 19, Air Conditioning Analysis.

Table 2 — Values of N and Θ

N	Θ	(90—Θ)	N	Θ	(90—Θ)	N	Θ	(90—Θ)
0.00	0°— 0′	90°— 0′	0.50	38°—58′	51°— 2′	1.20	62°—45′	27—15
.01	0 —56	89 — 4	.51	39 —31	50 —29	1.22	63 — 8	26—52
.02	1 —51	88 — 9	.52	40 — 4	49 —56	1.24	63 —30	26—30
.03	2 —47	87 —13	.53	40 —36	49 —24	1.26	63 —52	26— 8
.04	3 —42	86 —18	.54	41 — 8	48 —52	1.28	64 —13	25—47
0.05	4°—37′	85°—23′	0.55	41°—40′	48°—20′	1.30	64°—34′	25°—26′
.06	5 —33	84 —27	.56	42 —10	47 —50	1.32	64 —54	25— 6
.07	6 —28	83 —32	.57	42 —41	47 —19	1.34	65 —14	24—46
.08	7 —22	82 —38	.58	43 —10	46 —50	1.36	65 —33	24—27
.09	8 —17	81 —43	.59	43 —40	46 —20	1.38	65 —52	24— 8
0.10	9°—11′	80°—49′	0.60	44°— 9′	45°—51′	1.40	66°—10′	23°—50′
.11	10 — 5	79 —55	.61	44 —37	45 —23	1.42	66 —28	23—32
.12	10 —59	79 — 1	.62	45 — 5	44 —55	1.44	66 —46	23—14
.13	11 —53	78 — 7	.63	45 —32	44 —28	1.46	67 — 3	22—57
.14	12 —46	77 —14	.64	46 — 0	44 — 0	1.48	67 —20	22—40
0.15	13°—38′	76°—22′	0.65	46°—26′	43°—34′	1.50	67°—36′	22°—24′
.16	14 —31	75 —29	.66	46 —52	43 — 8	1.55	68 —15	21—45
.17	15 —23	74 —37	.67	47 —18	42 —42	1.60	68 —52	21— 8
.18	16 —14	73 —46	.68	47 —44	42 —16	1.65	69 —28	20—32
.19	17 — 5	72 —55	.69	48 — 8	41 —52	1.70	70 — 1	19—59
0.20	17°—56′	72°— 4′	0.70	48°—33′	41°—27′	1.75	70°—33′	19°—27′
.21	18 —46	71 —14	.71	48 —57	41 — 3	1.80	71 — 3	18—57
.22	19 —35	70 —25	.72	49 —21	40 —39	1.85	71 —31	18—29
.23	20 —24	69 —36	.73	49 —45	40 —15	1.90	71 —59	18— 1
.24	21 —13	68 —47	.74	50 — 7	39 —53	1.95	72 —25	17—35
0.25	22°— 1′	67°—59′	0.75	50°—30′	39°—30′	2.00	72°—49′	17°—11′
.26	22 —49	67 —11	.76	50 —52	39 — 8	2.10	73 —36	16—24
.27	23 —36	66 —24	.77	51 —14	38 —46	2.20	74 —18	15—42
.28	24 —22	65 —38	.78	51 —36	38 —24	2.30	74 —57	15— 3
.29	25 — 8	64 —52	.79	51 —57	38 — 3	2.40	75 —33	14—27
0.30	25°—53′	64°— 7′	0.80	52°—18′	37°—42′	2.50	76°— 7′	13°—53′
.31	26 —38	63 —22	.82	52 —59	37 — 1	2.60	76 —38	13—22
.32	27 —22	62 —38	.84	53 —39	36 —21	2.70	77 — 6	12—54
.33	28 — 6	61 —54	.86	54 —17	35 —43	2.80	77 —33	12—27
.34	28 —49	61 —11	.88	54 —55	35 — 5	2.90	77 —58	12— 2
0.35	29°—31′	60°—29′	0.90	55°—31′	34°—29′	3.00	78°—21′	11°—39′
.36	30 —13	59 —47	.92	56 — 6	33 —54	3.50	79 —59	10— 1
.37	30 —54	59 — 6	.94	56 —40	33 —20	4.00	81 —13	8—47
.38	31 —35	58 —25	.96	57 —13	32 —47	4.50	82 —11	7—49
.39	32 —15	57 —45	.98	57 —45	32 —15	5.00	82 —57	7— 3
0.40	32°—54′	57°— 6′	1.00	58°—17′	31°—43′	6.00	84°— 7′	5°—53′
.41	33 —33	56 —27	1.02	58 —47	31 —13	7.00	84 —57	5— 3
.42	34 —12	55 —48	1.04	59 —16	30 —44	8.00	85 —35	4—25
.43	34 —49	55 —11	1.06	59 —45	30 —15	9.00	86 — 4	3—56
.44	35 —26	54 —34	1.08	60 —13	29 —47	10.00	86 —28	3—32
0.45	36°— 3′	53°—57′	1.10	60°—40′	29°—20′	12.00	87°— 3′	2°—57′
.46	36 —39	53 —21	1.12	61 — 6	28 —54	14.00	87 —28	2—32
.47	37 —15	52 —45	1.14	61 —32	28 —28	16.00	87 —47	2—13
.48	37 —50	52 —10	1.16	61 —57	28 — 3	18.00	88 — 2	1—58
.49	38 —24	51 —36	1.18	62 —21	27 —39	Inf.	90 — 0	0— 0

Fig. 4—Diagrammatic illustration showing how spray water temperatures are determined on water-air chart



IF WE HADN'T EXPORTED SCRAP TO JAPAN WOULD IT HAVE MADE ANY DIFFERENCE IN THE AMOUNT OF MUNITIONS MADE BY JAPAN?

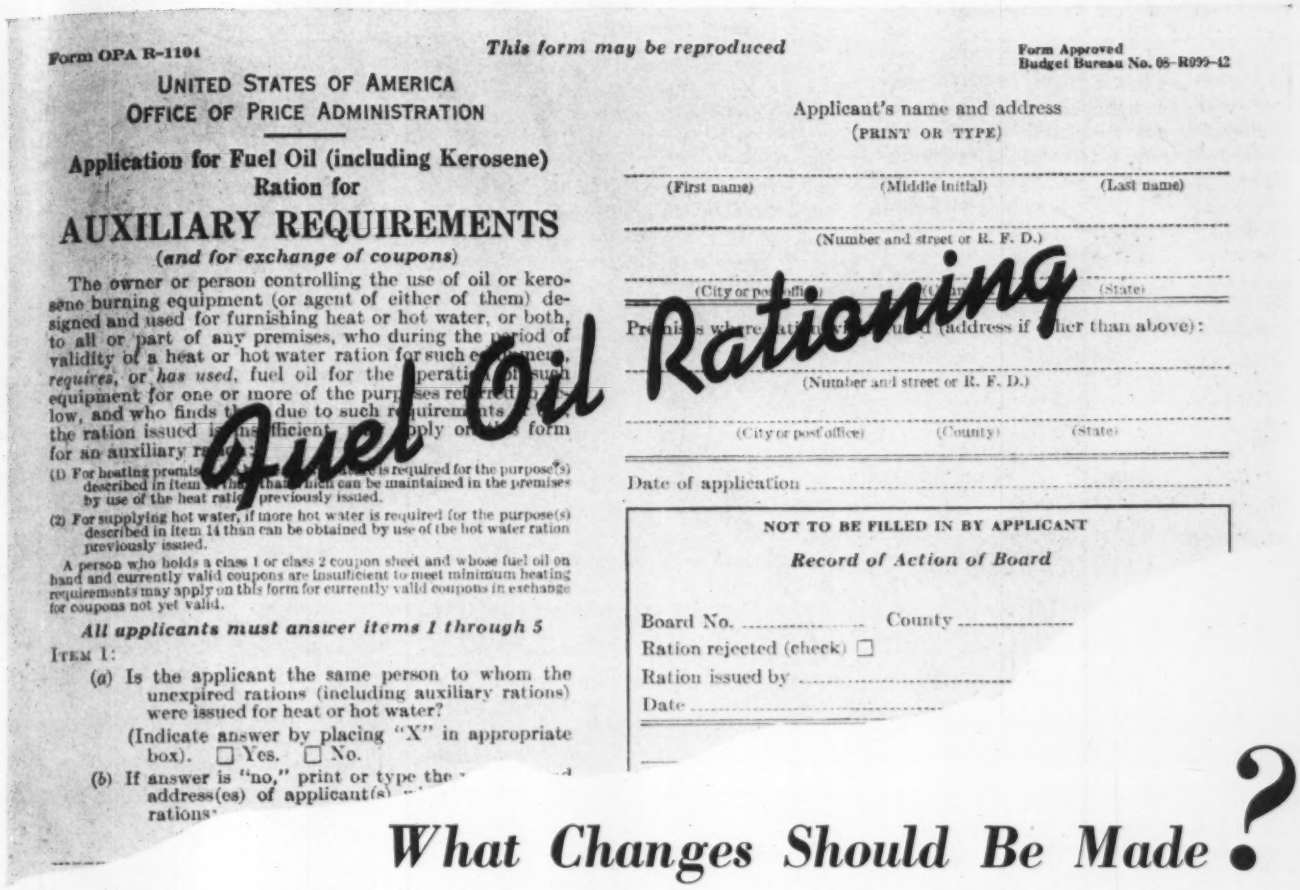
It is hardly conceivable that the comparably small amount of scrap sent to Japan would make or break Japan's armament program.

At our present rate of scrap consumption in the United States, the amount exported to Japan in 10 years would only last our furnaces a little over two months.

In fact, our shipment of scrap to Japan may actually be said to have injured her production capacity. For, had she not received these shipments, Japan would have built blast furnaces with which to manufacture steel from her ample pig iron in Manchukuo. Instead of that, she built open hearth furnaces which can no longer be supplied with scrap from the United States. Japan is, therefore, geared for scrap and being almost impoverished in that material, has been greatly embarrassed.—Business Press Industrial Scrap Committee, 3303 Empire State Bldg., New York, N. Y.

GET YOUR SCRAP IN THE FIGHT!

SALVAGE FOR VICTORY



Form OPA R-1104 *This form may be reproduced* Form Approved Budget Bureau No. 08-R099-42

UNITED STATES OF AMERICA
OFFICE OF PRICE ADMINISTRATION

Application for Fuel Oil (including Kerosene) Ration for

AUXILIARY REQUIREMENTS

(and for exchange of coupons)

The owner or person controlling the use of oil or kerosene burning equipment (or agent of either of them) designed and used for furnishing heat or hot water, or both, to all or part of any premises, who during the period of validity of a heat or hot water ration for such equipment *requires*, or *has used*, fuel oil for the operation of such equipment for one or more of the purposes referred to below, and who finds that due to such requirements the ration issued is insufficient, may apply on this form for an auxiliary ration:

(1) For heating premises, if more fuel oil is required for the purpose(s) described in item 14 than can be maintained in the premises by use of the heat ration previously issued.

(2) For supplying hot water, if more hot water is required for the purpose(s) described in item 14 than can be obtained by use of the hot water ration previously issued.

A person who holds a class 1 or class 2 coupon sheet and whose fuel oil on hand and currently valid coupons are insufficient to meet minimum heating requirements may apply on this form for currently valid coupons in exchange for coupons not yet valid.

All applicants must answer items 1 through 5

ITEM 1:

(a) Is the applicant the same person to whom the unexpired rations (including auxiliary rations) were issued for heat or hot water?

(Indicate answer by placing "X" in appropriate box). ☐ Yes. ☐ No.

(b) If answer is "no," print or type the name and address(es) of applicant(s) to whom unexpired rations were issued.

Applicant's name and address
(PRINT OR TYPE)

(First name) (Middle initial) (Last name)

(Number and street or R. F. D.)

(City or post office) (County) (State)

Premises where ration is used (address if other than above):

(Number and street or R. F. D.)

(City or post office) (County) (State)

Date of application

NOT TO BE FILLED IN BY APPLICANT

Record of Action of Board

Board No. County

Ration rejected (check) ☐

Ration issued by

Date

Fuel Oil Rationing

What Changes Should Be Made?

MEMBERS of the American Society of Heating and Ventilating Engineers, functioning through the society's national war service committee and similar committees set up by various local chapters, have done an outstanding job this heating season in assisting local rationing boards on problems involving fuel oil. In many cases, engineers have become regular board members, and in others they have served as consultants to such boards. . . . The editors recently asked some of those who have been particularly active in this work to comment on the present rationing scheme and any suggested improvements for next winter. The first three of the following communications are in answer to this request; they are supplemented by contributions and items from other sources, and further comment from readers is invited.

SUCCESS OF RATIONING REQUIRES ENGINEERS' AID

THE FUEL oil rationing regulations established and put into effect during the 1942-43 heating season involved problems for both oil users and distributors. The basic problem, however, was that of the ration—how much oil the consumer was to be allowed. This is the one that concerned and still concerns the millions of fuel oil users in the United States, the one that determines comfort, and the one that might well influence health.

The formula that was used last season in arriving at the amount of oil that would be allowed for heating was a very general one that depended for its accuracy upon the correctness of the assumption that there would be a reasonably uniform relationship between the floor area of a heated building and the heat loss from it. From the viewpoint of the heating engineer it was far from scientific and could fit only those cases that came under some typical class of construction and where exposed wall, roof, and window area bore a fixed relationship to the floor area. Since most of the heat loss from a building is through outside walls and roof, a more nearly correct formula could have been evolved using these measurements instead of floor areas. No doubt the user would have found it as simple to give those areas as to fill in the areas requested in the OPA form No. R-1100 as finally evolved. While this would have been an improvement, it still would be far from providing an exact measurement for heat loss.

However, a survey has shown that in a majority of the cases the rough floor area formula of 1942-43 has worked out satisfactorily. The unsatisfactory cases are those that fall far outside the "typical" classification, which includes among others, large homes with comparatively few occupants, high ceilings, one story buildings, and thermally inefficient homes that are not practical to reconstruct to bring into the thermally efficient class. For these, there must be modifications or amendments to the regulations permitting higher allotments of fuel if hardships and inequalities are to be avoided.

Much of the success of fuel oil rationing last winter can be attributed to the use of common sense by local boards in making adjustments in allowances that were not provided for in the regulations. OPA might well take cognizance of the steps taken by local boards in emergencies and make provisions for them through amendments, in order that there be uniformity of practice. Americans will bear many restrictions without complaint if they are necessary and if they are applied equally to all.

Rationing will be with us for at least another year. The basic ideas in the fuel oil rationing program last year will likely be retained next season in the interest of expediency. We have had assurances, however, that there will be amendments and orders modifying regulations to cover the hardship cases as completely as possible and to simplify deliveries. Engineers can now be of greatest help by assisting rationing boards in applying the regulations and orders in a sensible way. We must curb our natural impatience with the delays and uncertainties that come from so many apparently conflicting or overlapping government agencies that have control over some part of the fuel oil problem. Government is not efficient in business but it is now in business so we should do what we can to make its orders work during this war emergency. Hundreds of the members of

the American Society of Heating and Ventilating Engineers contributed valuable service in the fuel oil rationing program last season. Their services will be appreciated even more during the season ahead if fuel is so limited comfort can be sustained with difficulty, health jeopardized by a curtailment of ventilation below minimum accepted standards, and when the impact of the demands for conversion from liquid to solid fuels is felt. The program can be most successful only if engineering advice is followed. There should be more engineers in OPA.—JOHN HOWATT, chairman, engineering advisory committee for fuel conservation, Illinois chapter, ASHVE; member of HPAC's board of consulting & contributing editors.

ELIMINATE PERIOD COUPONS, INDISCRIMINATE CONVERSION

The necessity for oil rationing is twofold. First, the loss of tankers transporting oil to the Atlantic coast has created a tremendous transportation problem which is not yet solved. The 24 in. pipe line from Texas to southern Illinois helps the transportation and will help still more when completed to the Atlantic coast. If the storage facilities of the East can be filled to the brim this summer, these things together should ease the condition considerably as far as the Atlantic coast is concerned. There are other measures of relief in sight. The second factor is the tremendous war demand for our oil. The war must be fought and it must be won; and if it takes all of our oil, that is all right.

That does not, however, excuse in any way unnecessary rationing. It does not excuse trying to impose on districts where there is ample oil, the same rules and restrictions as applied to those districts where there is little oil.

Criticism has been made of the thumb rules set up in Washington for the rationing process. The application of engineering on such a scale was impossible; some sort of thumb rules were necessary, and any sort of thumb rules will be wrong a good deal of the time. Since, however, the thumb rule basis was necessary, it is also necessary that the local rationing boards must have the authority and the freedom to meet the situation as it exists in their local territory. That is the only way it can be carried out intelligently and beneficially.

There are many small users of oil, using from 2 to 10 gal a week perhaps, who have been put to great trouble and inconveniences to get their pittance of oil. The total amount is so inconsequential that a way should be devised to ease their burden. I suggest that they be given a card, instead of coupons, allowing them to purchase without restriction, except that of registering the purchase on the card, all the oil they need up to some limit to be established by the local board—a full year's supply.

The period coupon system has been a tremendous mistake and it has cost the oil distributing companies millions of dollars. The distribution problem, already severe on account of tires and gasoline restrictions, will be made worse as oil dealers find they cannot operate at a profit and quit the business, as several have told me they surely would. On account of the requirement for the deliveries of very small amounts of oil, the overhead of delivery has gone up anywhere from 100 to 500 per cent. Also the dealers are required to keep and balance a tremendous number of coupons — a headache in anyone's language. Relief should be given the dealers from both of these unnecessary and annoying costs.

Indiscriminate conversion has been a mistake. On the Atlantic coast, where there is ample coal and where the transportation of coal is for short distances, conversion of those major users of oil whose plants are capable of conversion should be forced at once; while similar plants located in the oil field should not be forced to convert.

In the Kansas City territory, the same attempts to force conversion are being made as on the Atlantic coast. Conversion in the Midwest, where the coal mining facilities are not sufficient to carry the whole load, means hauling coal from the eastern coal fields. Permitting the eastern industries to keep using oil, means hauling oil from the west to the east, which means double transportation.

Just as in the rationing itself, thumb rules of conversion will not meet the need. There must be intelligent engineering analysis of each district before conversion is required; and since a few large users use as much as many thousands of small users, conversion should start with the big users in the coal territory.

Rationing should be discontinued in territories where it has been definitely established that there has been no shortage of oil this past winter. Rationing under such circumstances is a detriment to the war effort and should not be continued.

The Midwest feels that it will do without anything and will do anything to help the war effort. The war must be won and the sooner the better, but things done in the name of the war effort, which interfere with the war effort, should either be modified or discontinued. Intelligence requires that.—E. K. CAMPBELL, E. K. Campbell Heating Co.

FUEL OIL TANKS TO BE FILLED DURING SUMMER

Modification of the present fuel oil rationing plan to achieve greater simplicity and aid the petrolum industry in meeting manpower problems was discussed at meetings held on March 16 and 17 between the Fuel Oil Industry Advisory Council and Office of Price Administration rationing officials.

The discussion included a review of the industry's experience during the past winter as well as the rationing method to be used next winter. Among the topics explored were possible alternatives to the use of coupons, improvements in the present coupon system, the relation between the industry's delivery problems and the five heating periods, as well as the method of issuing rations for the next heating season.

Joel Dean, chief of the OPA fuel oil rationing division, assured the industry representatives that every effort will be made to simplify further the rationing procedure and reduce the bookkeeping and clerical work of dealers and suppliers. A portion of the 1943-44 consumer rations, he stated, will be distributed in the early summer to enable the industry to fill householders' tanks during the period of small demand, provided primary supplies are at a high enough level to make this possible.

John Howatt and E. N. McDonnell, representing the American Society of Heating and Ventilating Engineers, attended the meetings.

FLAT PERCENTAGE CUT WOULD BE INEQUITABLE

Experience with apartments and student rooming houses leads me to the conclusion that they are being discriminated against by the application of a flat 33⅓ per cent cut. To all intents and purposes, these are homes just as much as private dwellings. A private dwelling that is well constructed and has been economical of oil is given the benefit of this in computing the ration. On the other hand, an apartment or rooming house of equally good construction is given no benefit. Under these circumstances they are helpless when a 33⅓ per cent cut is applied. They cannot do any more to conserve oil, and with this cut cannot maintain even 60 F. I think that some revision of this should be made in future rationing programs.

Owing to the lateness with which the rationing program was put into effect this year, it was not possible for most property owners to insulate or even install storm sash. Under these circumstances it is possible that a flat cut, based on the previous year's use, would have been advisable for private homes, too. But as a long range program it allows people who can get the most gain out of insulation and storm sash to live more comfortably than their neighbors who already had done everything possible to conserve oil before the cut was made.

The present scheme as applied to

homes admittedly has its weaknesses, but I have been unable to figure out one to replace it that would completely iron out all of the inequalities. Hence, I believe that it should probably be used as a basis for next year and some of the more obvious errors corrected.—A. P. KRATZ, research professor, University of Illinois.

START YEAR SEPTEMBER 1; ABANDON COUPONS

The current heating year, which began September 1, 1942, has been the coldest, in much of the United States, for the past 40 years. This has put a severe strain on the fuel oil rationing plan. Up to March 22, the records of the U. S. Weather Bureau in Chicago, show that 5762 degree days had accumulated, or 3 per cent more than for the entire heating year of 1941-42, and 26 per cent more than up to the same date a year ago. If the present proportion between the current year and the normal, or average, year continues to September 1, then the accumulation for the current year will be more than 7000 degree days, making this year the coldest year in the Chicago region since 1903-04. During these 40 years, only three have exceeded 6900 degree days, while 27 have registered less than 6300. Furthermore, the records show that in no case has an exceptionally cold year been followed directly by a mild year, while in several instances, an especially cold year has been followed by an even colder year. Therefore, it is reasonably safe to predict that the year 1943-44 will be another cold year with fuel requirements equal to those of this year.

Undoubtedly, most rationing boards have found it necessary to grant additional fuel oil in numerous cases (over and above the increase in value of 3rd, 4th, and 5th period coupons) to relieve actual or anticipated suffering. Before the end of next September, many oil users will have exhausted their quotas of oil. Up to March 22, only 82 per cent, or about five-sixths, of the heating season had been passed. Therefore, it will require one-fifth as much oil as a user had burned up to March 22, to carry him through to the end of September. Most users probably had less than this amount of oil or coupons on hand March 22.

The type of plan for rationing fuel oil which should be adopted by the OPA for next year is dependent upon whether fuel oil will be more or less available in different sections of the country than coal. So many contradictory statements have been published about the fuel oil and coal situations and the transportation limitations for coal and oil, that the public does not know what to believe. Everyone is willing to sacrifice comfort and convenience, and even endure real hardship, in order to help win the war; the user must be convinced that by so doing he is actually helping, and not being inconvenienced and made to suffer just because of unwarranted plans for rationing that will not materially aid the war effort.

I suggest the following modifications:

1) Change from October 1 to September 1 the date of the starting of the heating season, for September 1 is the date adopted by the U. S. Weather Bureau for all records of degree days (the measure of the accumulation of cold weather) and is used by heating and ventilating engineers for estimating purposes. Much confusion would be avoided by making this change and certainly in most of the country the heating season starts in September. Incidentally, if the OPA adopts September 1 for the start of its next rationing period, that would automatically help meet the problem of those users who will be unable to get through next September on the 1942-43 ration, because of the extraordinary coldness of this year.

2) After this year's experience with the coupon method of fuel oil rationing, it would seem unnecessary to continue this cumbersome and costly scheme another year. A simpler method that would take advantage of individual storage tank capacities and save many unnecessary small deliveries of oil would be to issue a card to each user for a full heating season's ration, based on some proportion (more or less) of what he actually used in 1942-43 (provided he was not definitely required to change over from oil to coal). This would enable him to purchase oil in such quantities and at such times as the capacity of his storage tank or tanks would permit. In cases of change of ownership or occupancy, simple rules could be used to determine the allotments for such cases. The ration card would apply to a definite person and a definite location, and would be taken up and reissued when a change of person or location occurred.

In any event, it is vitally essential that the OPA decide upon and announce a plan for rationing fuel oil for 1943-44 within the next 30 days. If less oil than was used in 1942-43 for heating will be available in parts of the country, it may be necessary to order thousands of oil users to change over from oil to coal burning. In that case, these users must be given adequate time to make necessary plans for coal burning equipment and for providing space for coal storage, as well as time for securing the equipment and labor for installing it. Owing to priorities, it may be next to impossible to purchase the equipment and with the increasing scarcity of labor it may be impossible to get competent labor to install what might be bought and to take care of hand fired equipment after it is installed. The problem is not a simple one and several months must be allowed for thousands of small and large users of fuel oil to change from oil to coal burning equipment.

It will not be surprising, it seems to me, if in some sections, where oil users have been compelled to change over to coal, that they may be ordered to convert back to oil burning, because of transportation difficulties attending delivery of coal, where oil can be more readily obtained.—THEODORE L. CONDRON, C. E.

ASKS SIMPLIFICATION OF RATIONING SCHEME

Drastic simplification of the method of fuel oil rationing was asked last month in a joint Illinois senate-house committee report. The committee comprised six Republicans and four Democrats under the chairmanship of Senator Earle Benjamin Searcy (Rep.). Three Democratic committee members did not sign the report.

The report asked elimination of the coupon system, allotment of rations to users regardless of the type of construction or equipment, cessation of forced conversions, encouraging the drilling of new wells, abolishment of multiple control over fuel oil, and abandonment of the premise that a simple system is not compatible with equitable distribution.

INDEX OF AMENDMENTS TO FUEL OIL RATIONING ORDER

(*From fuel oil industry letter No. 5, issued by Office of Price Administration.*)

Amendment 1, effective 10/28/42: Change of industry registration date.

Amendment 2, effective 10/31/42: Redemption date of coupon notes extended to 30 days after effective date of regulations.

British Ministry of War Shipping and other United Nations eligible on form OPA R-544 (revised).

Auxiliary heat rations for dentists, osteopaths, chiropractors.

Heat rations for railroad passenger cars on form OPA R-1102.

Board to which residents outside limitation area shall apply for ration.

*Consumer credit.

Transfers from within to without the limitation area to other than consumers.

Amendment 3, effective 11/14/42: Rations for space heaters in premises other than private dwelling.

Redemption of coupon notes by late registrants.

Transfers in absence of transferor; time period for forwarding coupon sheets in all absentee transfers.

False and misleading statements.

Class 3 coupon sheets.

*Date on which coupons must accompany transfers extended to November 23.

Amendment 4, effective 11/19/42: Computing rations for late applicants.

Applications executed prior to November 1 and filed not later than No-

vember 14 shall be treated as though made prior to November 1.

Transfers among dealers and suppliers on coupon credit.

Amendment 5, effective 11/23/42: Commingling of several dealers' supplies in the same storage facilities.

*Time for October report of primary suppliers extended to December 10.

Certification by heating contractor.

Amendment 6, effective 11/23/42: Appeals from board action.

Amendment 7, effective 11/13/42: Definition of "convertible facilities."

Interim ration for convertible facilities.

Review of rations granted for heat or hot water in premises other than private dwellings.

Applicant must prove equipment is not convertible.

Amendment 8, effective 11/14/42: Ration to be computed on basis of stated consumption where certification of purchases is unavailable.

Where hot water adjustment leaves minus quantity of "historical heat."

Space heater rations when certifications are not available.

Amendment 9, effective 11/25/42: Keeping of records by dealer with regard to certifications.

**Amendment 10*, effective 11/23/42: Extension of coupon credit.

Amendment 11, effective 12/10/42: Limitations on rations for space heaters. Trailer allowances increased.

Amendment 12, effective 11/30/42: Extension of coupon credit.

Amendment 13, effective 12/16/42: Extra coupons for late applicants on R-1102 and R-1103, to cover purchases from October 1 to date of application.

**Amendment 14*, effective 12/14/42: Extension of validity of period 1 and 2 coupons.

**Amendment 15*, effective 12/14/42: Extension of coupon credit.

Amendment 16, effective 12/26/42: Change in "passenger" definition.

Oil for coal spraying is limited. Identifying coupon sheets for auxiliary rations.

Forbidding consumption in excess of ration even though fuel oil in inventory.

Exchange of fuel oil within 24 hours. Permits consumption of fuel oil brought into area in supply tank of boat and locomotive.

**Amendment 17*, effective 12/18/42: Review of rations by boards on consumer's request.

Amendment 18, effective 12/24/42: *Advancing validity date of period 3 coupons.

Permitting suppliers to collect dealers' coupons within 5 days.

Change of unit value in 13 midwestern states.

Acceptance by supplier from board of exchange certificate of greater value than exchange certificate surrendered.

Submission by primary supplier of exchange certificate in excess of oil transferred.

Get Your Scrap in the Fight!

Salvage for Victory

**Amendment 19*, effective 12/28/42: Revocation of amendment No. 17.

Amendment 20, effective 1/8/43: Use of 1125 (delivery receipt) for large users of distillate oil for non-heating purposes.

**Amendment 21*, effective 1/4/43: Exchange of period 3 coupons at boards within 72 hours.

Amendment 22, effective 1/7/43: Redefinition of primary supplier.

Amendment 23, effective 1/7/43: 50 per cent reduction on 1101 rations for portion of ration used in heating non-residential space.

**Amendment 24*, effective 1/12/43: Coupon credit collections and cancellations.

Amendment 25, effective 1/9/43: Extension of time for exchange of period 3 coupons by primary suppliers.

Amendment 26, effective 1/9/43: Non-discrimination by secondary or primary suppliers.

Amendment 27, effective 1/20/43: 15 days before and after issuance of further ration.

Determination of further ration.

Amendment 28, effective 1/20/43: Heating rations of 200 gal or less in form of class 3 coupons.

Religious practitioners allowed to certify to need for auxiliary rations.

Inventory deductions made from currently valid coupons.

FBI, allowed special rations in certain cases.

Amendment 29, effective 1/18/43: 40 per cent cut on commercial, industrial, and governmental users not on list A of petroleum distribution order No. 3 by PAW.

Amendment 30, effective 2/1/43: Emergency facilities. Emergency deliveries. "Exchange" on coupon sheets. (Rations for emergency operation of fuel oil burning facilities when others are inadequate. Dealers allowed to deliver up to 50 gal without receiving coupons in special cases. Boards are to write "exchange" on coupons which they issue in exchange for future valid coupons.)

Amendment 31, effective 2/2/43: Trailer, malicious destruction, inventory coupons. (Trailer ration to be double maximum or amount requested, whichever is lower. Rations are to be revoked for person maliciously destroying grates, etc. Dealer may write name and address across several inventory coupons rather than insert registration number on each coupon. Delivery receipts may be used for heating purposes for residual oil and for distillate, if 100,000 gal or more monthly consumption.)

Amendment 32, effective 1/26/43: Deadwood. (All dead language, i.e., expired provisions, is removed.)

Amendment 33, effective 1/27/43: Emergency authority may be exercised by regional administrator.

Amendment 34, effective 1/28/43: Validity dates on period 4 coupons.

Amendment 35, effective 2/10/43: Cutoff date for class 1 and class 2 coupons.

Use of R-120 by dealers optional with receiving supplier.

Amendment 36, effective 2/1/43: Washington and Oregon included in rationed area.

Amendment 37, effective 2/9/43: Washington and Oregon—base period changed to nearest subsequent 12 months for which dealer has records.

Amendment 38, effective 2/23/43: Pleasure boating prohibited except certain specified cases.

Amendment 39, effective 3/2/43: Additional facilities definition changed to include stoves obtained under ration order 9 and advance date July 31, 1942, to December 19, 1942.

Amendment 40, effective 2/27/43: Regional administrator authorized to extend coupon credit period.

Amendment 41, effective 2/27/43: Extension of coupon credit—Washington and Oregon.

Amendment 42, effective 3/10/43: Increase to six months heating period for non-heating farm uses.

Amendment 43, effective 3/11/43: Heating expert's certification may be waived by boards in special cases.

Amendment 44, effective 3/2/43: Procedure for issuance of suspension order.

Amendment 45, effective 3/13/43: Allowance of extra rations to consumers who can prove general hardship under current rations.

Amendment 46, effective 3/13/43: Allowance of substitute rations for certain consumers who have had a change of circumstances since original application.

Amendment 47, effective 3/15/43: Issuance of stove certificate in place of 100 gal or more auxiliary ration unless applicant proves inability.

Amendment 48, effective 3/6/43: Advance of validity dates period 5 coupons.

Amendment 49. Adds 20 westernmost counties in Idaho to rationed area.

Amendment 50. Applicable to PAW district 1. Provides for adjustment of rations to certain consumers using fuel oil for electric generation.

*Obsolete.

"Open for Discussion"

DON'T RUN CONDENSATE FROM ROOM COOLERS DOWN BUILDING WALL

Handling Small Water Quantities with the Siphon or Ejector Pump

THE EDITOR—

I am writing relative to the article, Handling Small Water Quantities with the Siphon or Ejector Pump, by F. F. Stevenson, in the March HPAC. Reference begins with the last full paragraph in column one on p. 138. My comments:

Certainly it is appropriate in ground floor offices of industrial firms "to cut the wall, run a ⅜ in. condensate line through, and let the condensate help water the grass outside." That is undoubtedly an appropriate manner of handling this little matter for such conditions. I question seriously, however, the author's solution of the same problem in the case of coolers in modern office buildings dealt with in the next paragraph. Here again the author proposes that this little condensate line "be run through the exterior wall, the tubing curved a little to direct the water flow so that it impinges squarely on the wall surface, and the water simply trickles down the wall. Two or three floors down, there will be no sign of moisture. Of course, permission from the proper authorities should be obtained before proceeding with such work."

This is so much against the principles of good judgment and permissible practice that I think it should be ruled out as a solution for condensate disposal.

Within the experience of the undersigned, and some others with whom I have talked, the procedure would be ruled out automatically—for permission from the proper authorities (the managers or agents of the building) would definitely not be forthcoming.

A few years ago one of the tenants in one of our buildings purchased a room cooler and the installer piped the condensate through the outside wall to let it drip down the wall of the building. On a humid day this little ¾-ton job caused a fan shaped smear of water which extended down many stories; it was dripping on the window sills of other tenants below his floor and spattering down on the sidewalk—a completely intolerable condition. The tenant, not having obtained permission from the proper authorities to make such an installation, was required forthwith to have the installation corrected and the condensate piped into the proper channels provided for it (the building waste lines).

Contemplate for a moment the practical aspects of this manner of condensate disposal. If the water from each condensate pipe trickles down the wall "two or three floors," imagine a condition in a 15 or 20 story building where a number of tenants disposed in a vertical plane each have piped through the wall a small stream of water sufficient to "help water the grass outside." What would the side of the building look like? Again, if this were not merely one vertical tier, but if the offices were pretty generally equipped with these room coolers dripping water through the outside walls, what kind of a condition would you have on the outside of your building and what would your own tenants, and the public passersby who use your sidewalk, think of your building as an institution and of your office as managers? How would you placate your first floor store tenants, with this unsightly dripping down on the sidewalk where their potential customers should be able to inspect the merchandise displayed in their windows without resorting to the use of raincoats and umbrellas?

The quantity of water would be far from negligible. A 1 ton room cooler adequate for a fairly large single office has a Btu capacity of about 12,000 per hr. In humid summer weather as much as half of this capacity may be utilized in dehumidifying the air which it is handling. This would be 6000 Btu per hr. Allowing approximately 1000 Btu for each pound of vapor condensed into water, this would mean 6 lb or approximately ¾ gal per hr of water per unit, and this is not an inconsiderable amount even when contemplating a single installation.

Since individual room coolers are one solution of the engineering problem of cooling offices in public office buildings and since, therefore, it may be expected that there will be more and more of them in use, it is unthinkable for the air cooler trade to contemplate disposing of the water of condensation by dripping it down the outside of the landlord's building. The problem has to be met in some other manner more acceptable to the owners and managers of these costly investment structures, and less offensive to the people who look at them, use them, and go past them on the sidewalks below.

One means, not of solving the condensate problem but of avoiding it, is the use of the type of cooler which re-evaporates and returns to the outside atmosphere the moisture removed from the conditioned air. This, of course, does not answer the problem of the manufacturer who is endeavoring to sell the units which remove the moisture as liquid condensate and that is still their problem. No building manager who merits the title is going to let his building be used as an outdoor spray tower and if he were inclined to permit such practice even his best tenants would move out of his building in disgust, regardless of whether they themselves had contributed to this intolerable condition.

The author, Mr. Stevenson, proposes the real answer to the condensate question—a pump.—R. E. HIERONYMUS, general superintendent, Aldis & Co., member of HPAC's board of consulting & contributing editors.

REPLY—

Mr. Hieronymus, in general, is right. The picture he paints is enough to cause any building manager to turn pale and tremble.

All levity aside, my article suggested this method only in an emergency. However, I have seen several installations with none of the devastating effects described by Mr. Hieronymus.

I believe Mr. Hieronymus overemphasizes the amount of condensate. The average cooler has a total capacity of 9000 to 10,000 Btu per hr, and even on humid days, there is seldom more than 3000 to 4000 Btu of latent heat removal. But I still think Mr. Hieronymus is right.—F. F. STEVENSON.

If the heat release rate is too high, as evidenced by a marked tendency of the refractories to fusion, it may be necessary to provide more furnace volume by construction of a dutch oven in front of the boiler's main firebox

★

Boiler Firebox Maintenance

Kalman Steiner Discusses Refractories and the Importance of Their Proper Use Under Wartime Conditions in Plants and in Buildings

SUMMARY—Good management calls for proper maintenance of boiler refractories in normal times; in wartime, with plants operating on heavy schedules and carrying overloads, refractories are put to even more severe use, and the problem becomes more important. There is also the possibility that a few changes in the design or construction of a combustion chamber may result in worthwhile—and patriotic—fuel savings . . . Mr. Steiner, member of HPAC's board of consulting & contributing editors, a well-known authority on combustion engineering who is now engaged in ordnance work, discusses behavior of refractories under service conditions and chief causes of failure. The cardinal factor in firebox maintenance is alertness to detect the first signs of deterioration . . . An earlier article by Mr. Steiner in the October, 1942, HPAC described the nature of refractories, their properties, and principal applications

IT IS important for the operating engineer to know the limitations of the materials with which boiler settings are constructed and maintained, and the manner in which they are affected by use. It is convenient to classify service effects according to the type of failure that can be encountered. These are usually termed spalling, fusion, and slagging.

Spalling means cracking or shattering of the brick to such an extent that pieces break away and thereby expose fresh surface to the furnace conditions. It is probably the most general class of firebrick failure. There have been three sorts of spalling recognized. The first is thermal, and results from expansion and contraction due to temperature change—the more rapid the change the more severe the thermal shock. Brick having the lowest coefficient of expansion, the most uniform rate of expansion, and the greatest elasticity will have the best resistance to thermal spalling. Superduty firebricks are especially resistant to it. The second type of spalling is that caused by mechanical abrasion or pressure. Manipulation of firing tools to break away accumulations of clinker, or pinching of firebrick by expansion, are examples of what can induce mechanical spalling. The third class of spalling is internal to the brick, and consists in structural alteration of the brick's composition under the influence of chemical or physical environment, usually making the brick more brittle.

In order to understand the phenomena of *fusion* in firebrick, it is best to visualize the materials and forces present in a hot firebox. Firebrick is not a single chemical substance, but a mixture of numerous substances. Some bricks will have melting points lower than others. As a brick enters that region of temperature known as its softening range, the lower melting particles liquefy into viscous masses distributed throughout the brick. If there is no external pressure or load applied, the brick will retain its orig-

inal shape until practically the entire solid has melted. But under load, the liquid acts as a lubricant and permits the solid portions to slip over one another, and so the brick is deformed. The rate of deformation will depend upon the composition of the brick, the temperature, and the intensity of the load.

By *slagging* is meant the chemical reaction of refractories with other substances. In the case of boiler fireboxes, the reacting foreign substances are chiefly ash and gases. Fireclay materials are basic; that is, they are chemically on the alkali side of neutrality. Ashes and combustion gases are acidic, sulphur dioxide much more so than carbon dioxide. Basic and acidic substances have a marked affinity for one another, and they combine chemically to form neutral salts known as slag, which is the familiar glassy appearing refuse of metallurgical industries. Slagging is seldom encountered in oil fired boilers unless moist combustion gases are allowed to accumulate in cold settings. The extent to which coal ash forms slag with refractories depends principally upon the melting point of the ash and the extent of the area of contact. Ash with high melting points may adhere to and build up sizeable accumulations on the firebox walls, but the chemical union is slight and the only ill result will be mechanical abrasion in careless removal of the cinder. Molten ash flows over the firebox surface and enters into a chemical action that forms brittle glass or slag. Coal burned on stokers and grates will affect the firebox only at the grate line unless high gas velocities carry small particles upward. Ash from pulverized coal can come in contact with much more of the setting.

Watch for Deterioration

How can the operating engineer apply these facts about refractories, their properties, and behavior to the preservation of existing fireboxes and the conservation of these valuable materials? The cardinal factor is alertness to detect first signs of deterioration in the firebox. Just as a single rotten apple will quickly cause the entire barrel to spoil, so the first brick that fails in a wall will allow forces to act upon the surrounding bricks and rapidly induce damage throughout. When a brick spalls by the breaking away of a piece, not only is a new vertical surface of that brick exposed, but so also are the horizontal faces of the bricks above and below. To check disintegration, therefore, a broken brick should carefully be cut out and replaced by a new one accurately fitted into place. When erosion is deep in a small area, the hole may be cleaned out and plugged with plastic firebrick. When the affected area is of wide extent but not deep, it is better to repair with new brick.

A rather common occurrence around port holes is the cracking of plastic cones, allowing pieces to break away and expose metal parts to firebox temperatures. Unless repairs are made promptly the metal will be warped or cracked. Repeated failures of plastic in such spots are an indication that the design should be remodeled to make the plastic heavier or to reinforce it with iron prongs. If the cracking results from frequent need to remove carbon deposits, the combustion conditions should be studied with a view to preventing the carbon formations.

A common construction method in boiler fireboxes is to spring an arch to hold up a wall through which combustion apparatus is inserted. If such an arch is not securely anchored at the ends, it soon deteriorates. When stokers and oil or gas burners are installed, firebrick liners are frequently used inside the main furnace walls. If an arch is sprung from such a liner to the opposite one the foundation may be so insecure that the arch soon crumbles. The remedy is to support the arch from the main walls, and proper skewbacks must be provided upon which the arch may rest firmly. If the arch is subject to extreme temperature and fusion is rapid, then a relieving arch should be built above the main arch, to furnish support for the main wall above. An arch will not fail from temperature alone nearly as rapidly if it does not also have to support the weight of the overlying brickwork. Whenever the combustion arch must be rebuilt, the relieving arch by supporting the main portion of the wall facilitates the work.

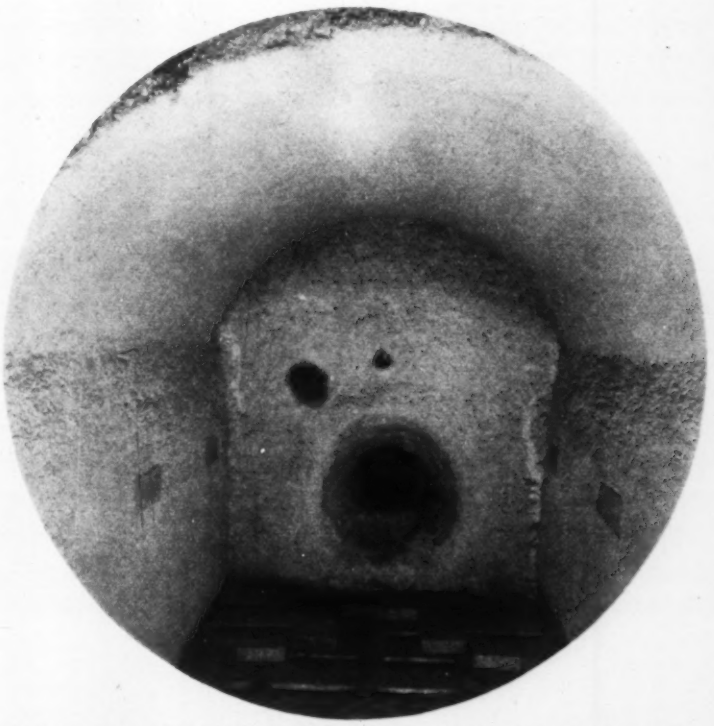
Check Heat Release Rate

When furnace refractories have a marked tendency toward fusion, as is usually manifested by a glassy coating over the brickwork and perhaps with pools of fused material collected at the bottoms, the furnace volume should be checked for rate of heat release. With standard first quality or high heat brick it is not considered advisable to exceed a maximum heat release of 30,000 Btu per cu ft, although there are many furnaces in use where the rate runs as much as 50 per cent greater. Should the calculation of heat release show a rate above 30,000 in a furnace exhibiting marked tendency toward fusion, then certainly steps should be taken to provide more combustion volume, either by excavating and dropping the floor or by erection of a dutch oven in front of the main firebox.

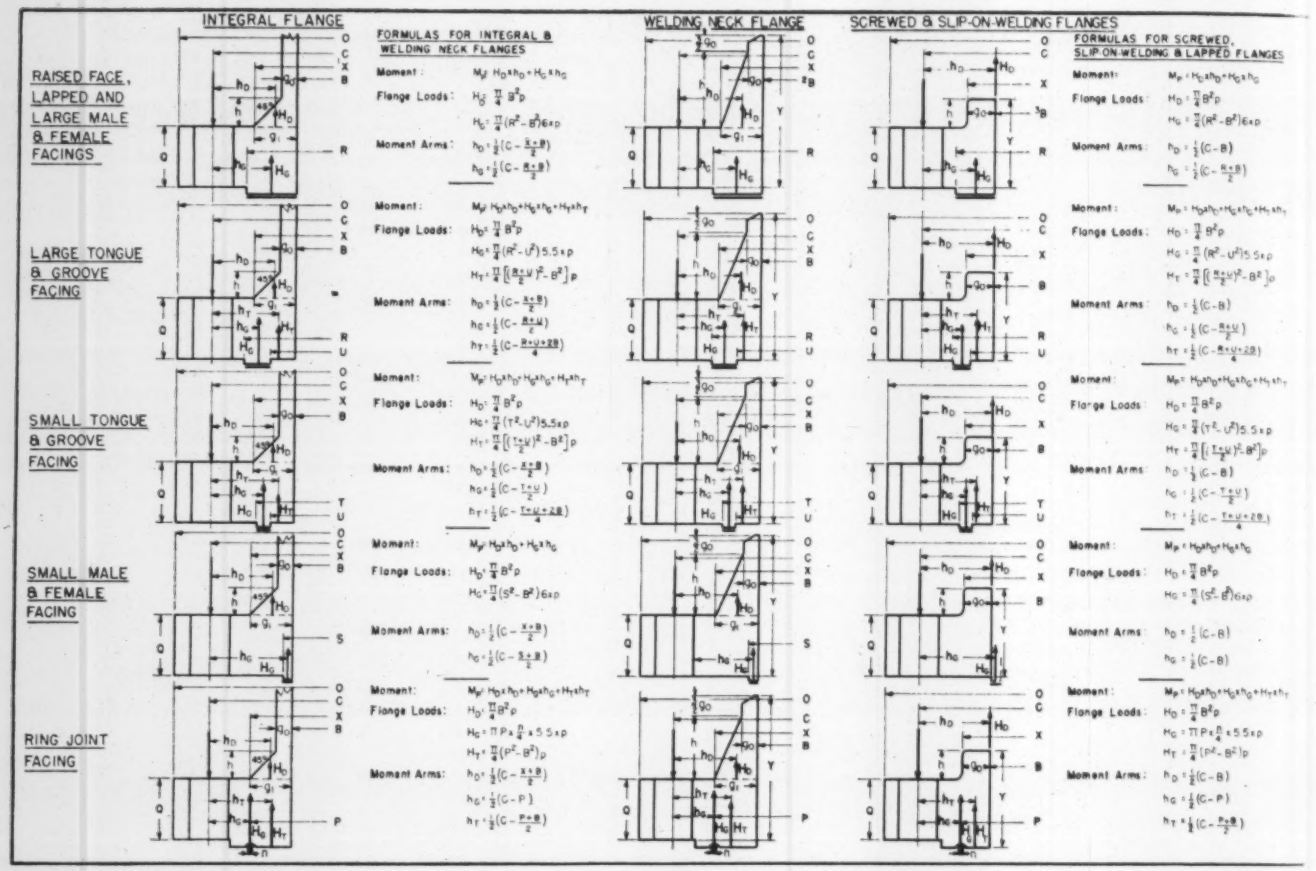
Sometimes only the floor of a firebox attains fusion temperature, and when there are air ports in such a floor the running of the refractory may close the port. This kind of floor usually is of a checkered construction, and the floor can be dropped by eliminating the checkerwork and introducing secondary air directly under the burners. The increased combustion volume and the increased distance from flame to floor will both be helpful.

Finally, in the case of a front fired oil or gas burner on a boiler with a short firebox, consideration should be given to reinstalling the burner for rear firing, which will give a longer flame travel and permit extending the firebox length backward.

In all cases where refractories are repaired or replaced, it is quite important that the correct materials be selected to meet the specific conditions prevailing.



Interior view of dutch oven, made of plastic with checker floor of firebrick. (Both photos courtesy A. P. Green Fire Brick Co.)


INTEGRAL FLANGE
WELDING NECK FLANGE
SCREWED & SLIP-ON-WELDING FLANGES
FORMULAS FOR INTEGRAL & WELDING NECK FLANGES
FORMULAS FOR SCREWED, SLIP-ON-WELDING & LAPPED FLANGES
RAISED FACE, LAPPED AND LARGE MALE & FEMALE FACINGS
LARGE TONGUE & GROOVE FACING
SMALL TONGUE & GROOVE FACING
SMALL MALE & FEMALE FACING
RING JOINT FACING

WAR STANDARD FOR FLANGES

N. O. Smith-Petersen Describes the ASA's 1943 War Standard for Pressure-Temperature Ratings of Steel Flanges, Flanged Fittings, and Valves

SUMMARY—Standard pressure ratings for steel flanges and fittings have been available since 1927, and have been revised and extended in 1930, 1939, and this year. The 1943 revision comprises a further effort toward keeping the standard abreast of industrial and engineering progress, and furthers economy in the use of steel . . . Mr. Smith-Petersen describes here the procedure and calculation methods used for the work of revision; he is design engineer, the Walworth Co., and chairman of the ASA committee which formulated the revision

STANDARD PRESSURE ratings for steel flanges and fittings have been in existence since 1927, in which year the American Engineering Standards Committee published its tentative American Standard B16e, entitled Steel Pipe Flanges and Flanged Fittings. This standard has been revised and extended to meet the needs of the industry, revisions being published in 1930 and 1939. The 1943 revision is a further effort toward keeping the standard abreast of industrial progress and engineering development.

The task of revising the 1939 standard was originally delegated to a special subgroup of Sectional Committee B16, on Standardization of Pipe Flanges and Fittings, of the American Standards Association. This committee's status, however, was later changed to the status of an ASA War Committee to facilitate the completion of this revision.

The special subgroup began its work previous to our country's entry into the war, and had a comprehensive plan for ascertaining strength of flanges as well as a plan to extend the pressure-temperature ratings to a greater number of flange materials. From the preliminary work on these plans it became apparent that the pressure-temperature ratings could be increased. War requirements for steel appeared to exceed our steel producing capacity, and economy in the use of steel became a national demand. Higher pressure-temperature ratings for steel flanges, valves, and fittings would further this economy in the use of steel, and the special subgroup took steps to hasten the completion of their rating revision. The comprehensive plan was discarded and an abbreviated plan for the strength analysis of the flanges was adopted. Thereby, instead of making approximately 2400 flange calculations, only approximately 450 flange calculations were required. This was accomplished by introducing the use of "probing" calculations in place of complete calculations. The probing calculations consisted of the calculation of the flange strength for the 900 psi pressure series for all the different flange types, flange facings, and flange sizes. In addition to this, another set of probing calculations was made consisting of the calculation of the flange strength for one flange size for all pressure series.

Calculations of the flange strength of the 900 psi pressure series disclosed the effect of flange facings upon the strength of the flange as well as the difference in strength of the various flange types. These calculations also gave information concerning the difference in strength between small and large flanges. Calculations of one flange size for all pressure series disclosed the strength interrelation between pressure series. Having obtained this information about the strength of flanges in general, the unprobed (or not calculated) flanges were assumed to follow the same

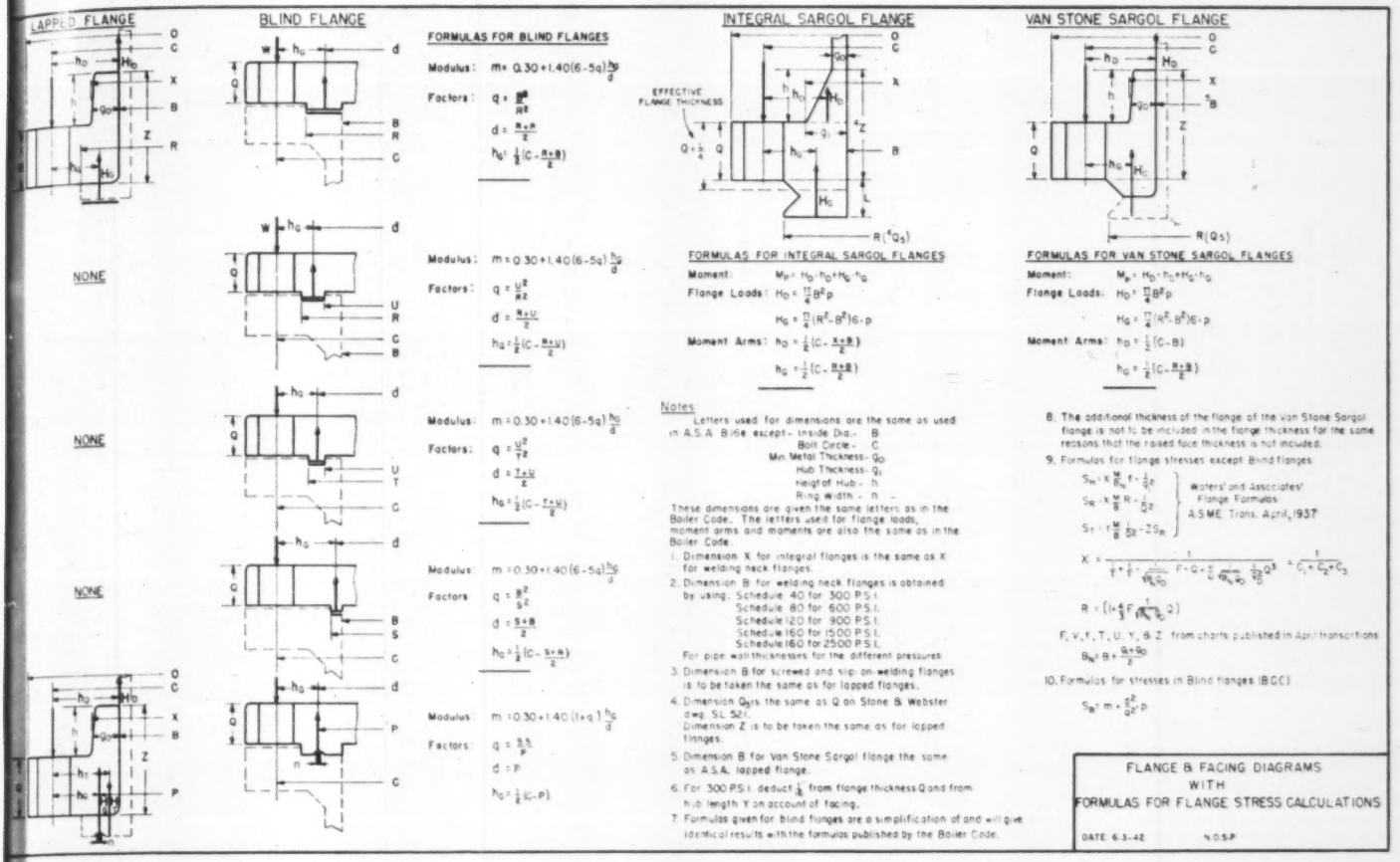


Fig. 1—Flange and facing diagrams with formulas for flange stress calculations

strength pattern. Upon this basis of calculation the revised pressure-temperature ratings were determined.

To accomplish the work of making approximately 450 flange strength calculations, each member of the special subgroup arranged with the engineering department of the company he represented to undertake a certain portion of it. For this purpose there were then prepared a special formula sheet and printed forms for calculation procedure in order to facilitate calculations and insure correct results.

For the flange strength calculations the Waters' and associates' flange formula was used. This is the only flange formula which permits calculations of stresses in three axial directions, and the formula has been approved by the ASME Boiler Code Committee. The flange stress formulas and diagrams showing the location of flange loads, with formulas for their amounts and moments for all the different facings and flange types, are shown in Fig. 1.

For use in connection with flange strength calculations the ASME Boiler Construction Code gives valuable gasket data. From these data a "gasket factor" of 5.5 was selected for the flange calculations. In addition to the flange loading prescribed by the ASME Boiler Construction Code a pipe line expansion force was included. This force was taken as equivalent to a flange bolt stress of 2000 psi.

The calculations gave different stress results for the different pressure series, flange sizes, and flange types. If the stress results are used to indicate the order of strength for the pressure series, this order will be as follows: 300 psi, 2500 psi, 600 psi, 1500 psi, 400 psi, and 900 psi. For flange sizes the relative order of strength is: the small sizes first, the larger sizes next, and then the intermediate sizes. For flange types the integral flange is the strongest, the blind flange the least strong.

The stress results were separated in two groups, those for ring joint facing, and those for other than ring joint facing, and an average or compromise stress was determined for each group. This stress was then used to find the primary service temperature for the group from allowable stress-temperature curves by accepting as a primary service temperature the nearest 50 F temperature multiple below the actual temperature obtained by the stress. By taking the nearest 50 F temperature multiple below that actually determined, considerable conservatism in pressure rating was introduced, for the stress-temperature curve near the primary service temperature varies rapidly with the temperature. The primary service temperature was determined for two materials only, the same materials as are rated in the B16e-1939 standard, namely, carbon steel and carbon molybdenum steel. Both these steels are given the same rating for forgings and castings, just as in the 1939 standard, basing the rating upon the strength of the cast material.

The next step in the pressure rating of the flanges was to determine upon pressure ratings above and below primary service temperature. The procedure here followed was to make the pressure rating curves approach the allowable stress-temperature curves more closely than do the 1939 pressure rating curves. The B16e-1939 pressure-temperature rating is practically a straight line from the primary service temperature to room temperature. The allowable stress-temperature curve, however, is a curve with a decided vertical convex terminating in a maximum point at 650 F, from which point it runs horizontal to room temperature. There is, therefore, quite an area useful for rating purposes between the B16e-1939 pressure-temperature rating straight line and the allowable stress-temperature curve. It is within this area that the new pressure-temperature rating curve lies. Both the 1939 and the 1943 pressure rating curves for carbon steel and carbon molybdenum steel are shown on the charts reproduced on the next page.

The 1943 American War Standard on Pressure-Temperature Ratings for Steel Pipe Flanges, Flanged Fittings, and Valves, approved January 26, 1943, by the American Standards Association, gives detailed information concerning calculation results and materials, and also contains four tables of pressure-temperature ratings with charts. It is obtainable from the ASA, 29 W. 39th St., New York City, for 25c. It is designated B16e5-1943.

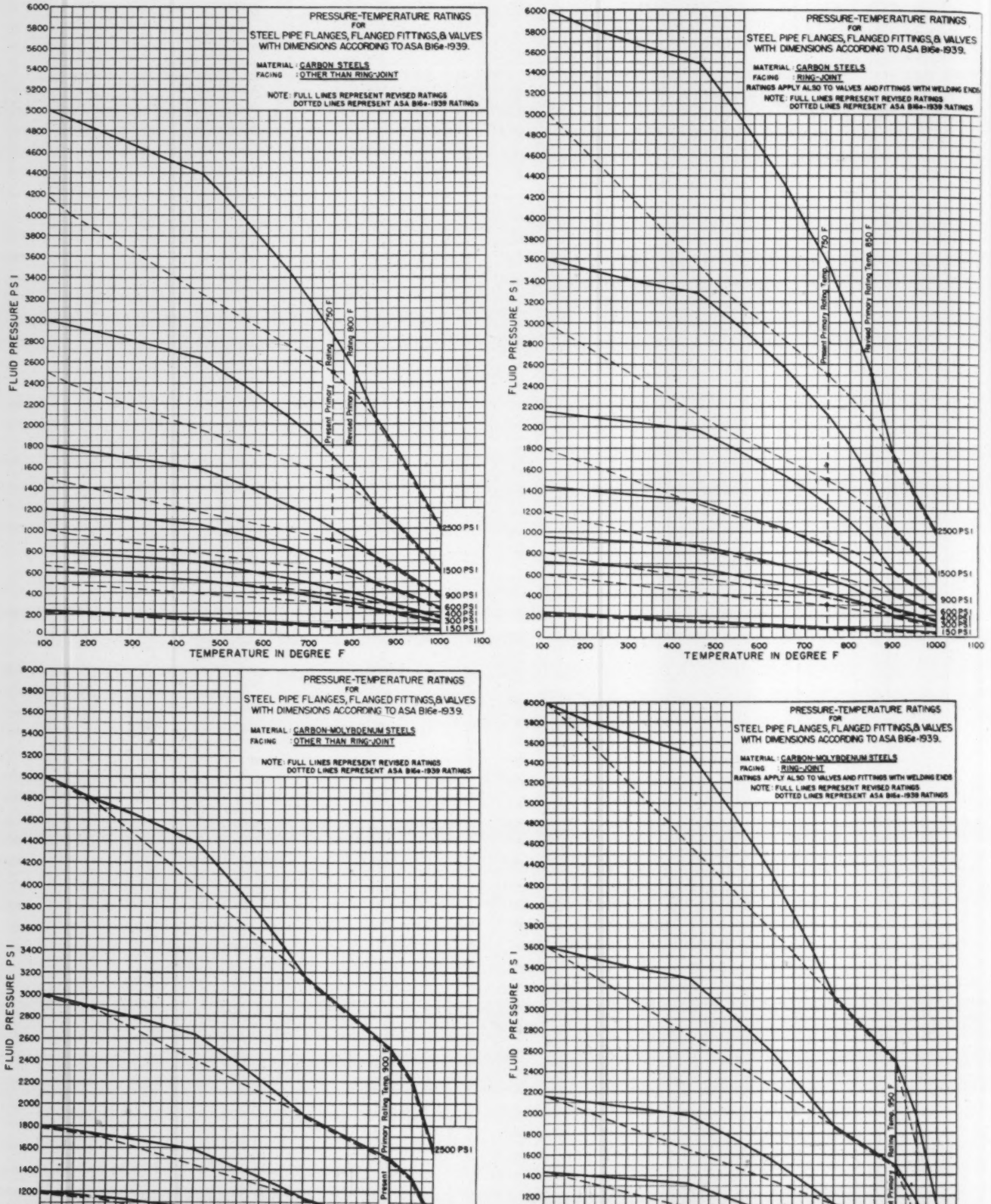
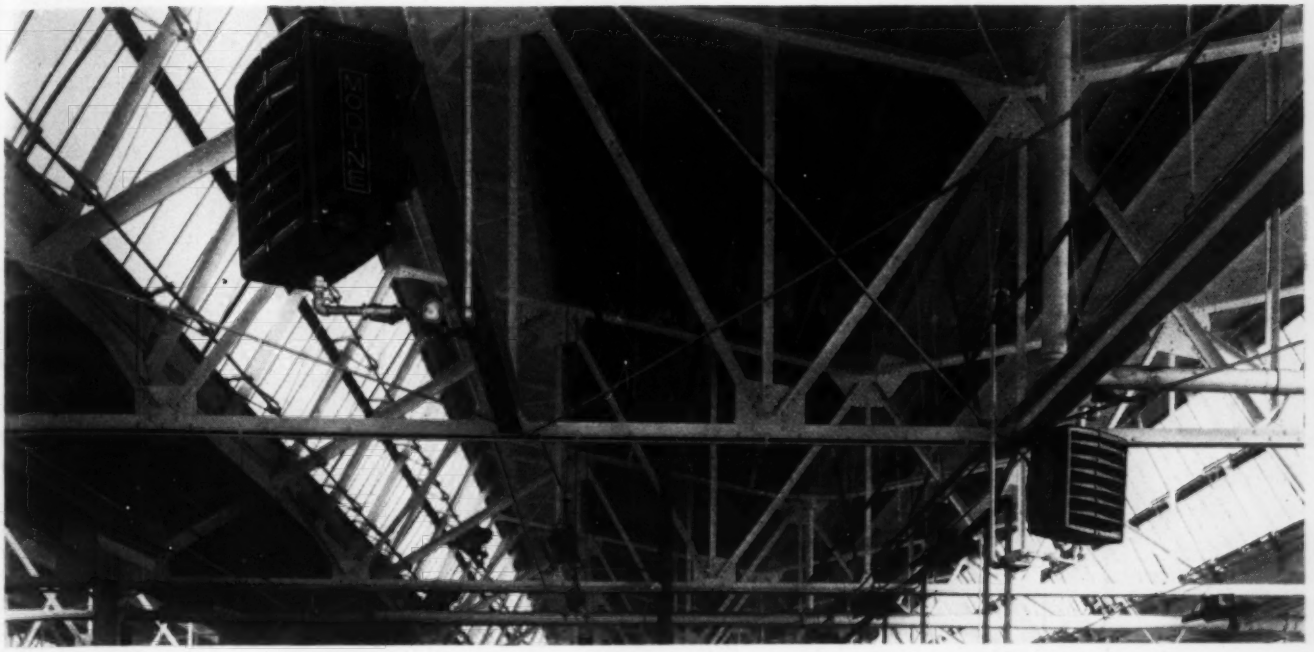

PRESSURE-TEMPERATURE RATINGS
FOR
STEEL PIPE FLANGES, FLANGED FITTINGS, & VALVES
WITH DIMENSIONS ACCORDING TO ASA B16e-1939.
MATERIAL : CARBON STEELS
FACING : OTHER THAN RING-JOINT
NOTE: FULL LINES REPRESENT REVISED RATINGS
DOTTED LINES REPRESENT ASA B16e-1939 RATINGS
FLUID PRESSURE P S I
TEMPERATURE IN DEGREE F
Present Primary Rating 750 F
Revised Primary Rating 800 F
2500 P S I
1500 P S I
900 P S I
600 P S I
400 P S I
300 P S I
150 P S I
PRESSURE-TEMPERATURE RATINGS
FOR
STEEL PIPE FLANGES, FLANGED FITTINGS, & VALVES
WITH DIMENSIONS ACCORDING TO ASA B16e-1939.
MATERIAL : CARBON STEELS
FACING : RING-JOINT
RATINGS APPLY ALSO TO VALVES AND FITTINGS WITH WELDING ENDS
NOTE: FULL LINES REPRESENT REVISED RATINGS
DOTTED LINES REPRESENT ASA B16e-1939 RATINGS
FLUID PRESSURE P S I
TEMPERATURE IN DEGREE F
Present Primary Rating Temp. 750 F
Revised Primary Rating Temp. 850 F
2500 P S I
1500 P S I
900 P S I
600 P S I
400 P S I
300 P S I
150 P S I
PRESSURE-TEMPERATURE RATINGS
FOR
STEEL PIPE FLANGES, FLANGED FITTINGS, & VALVES
WITH DIMENSIONS ACCORDING TO ASA B16e-1939.
MATERIAL : CARBON-MOLYBDENUM STEELS
FACING : OTHER THAN RING-JOINT
NOTE: FULL LINES REPRESENT REVISED RATINGS
DOTTED LINES REPRESENT ASA B16e-1939 RATINGS
FLUID PRESSURE P S I
TEMPERATURE IN DEGREE F
Present Primary Rating Temp. 900 F
2500 P S I
1500 P S I
900 P S I
600 P S I
400 P S I
300 P S I
PRESSURE-TEMPERATURE RATINGS
FOR
STEEL PIPE FLANGES, FLANGED FITTINGS, & VALVES
WITH DIMENSIONS ACCORDING TO ASA B16e-1939.
MATERIAL : CARBON-MOLYBDENUM STEELS
FACING : RING-JOINT
RATINGS APPLY ALSO TO VALVES AND FITTINGS WITH WELDING ENDS
NOTE: FULL LINES REPRESENT REVISED RATINGS
DOTTED LINES REPRESENT ASA B16e-1939 RATINGS
FLUID PRESSURE P S I
TEMPERATURE IN DEGREE F
Present Primary Rating Temp. 950 F
2500 P S I
1500 P S I
900 P S I
600 P S I
400 P S I
300 P S I

Fig. 2—Pressure-temperature ratings for steel pipe flanges, flanged fittings, and valves, with dimensions according to the ASA B16e—1939 standard, showing both the 1939 and the 1943 values. (See preceding two pages for explanation of the new war standard for flanges)



UNIT HEATER MAINTENANCE

Unit heaters, like any other mechanical equipment, will give trouble free service to the extent that they are properly cared for. Observance of a few simple rules will insure their satisfactory operation . . . prevent their damage and deterioration

Vital Points: Motor, condenser and casing.

Inspection: Every four months under average conditions; more often where air is contaminated with corrosive fumes, lint, dust, soot, or oil spray.

Motor

A) Cleaning

Remove grease and dirt on outside of motor during each inspection or lubrication. Motor should be dismantled every third year and cleaned thoroughly inside and out.

B) Lubrication

1) Ball Bearing Motors — Motors are shipped packed with lubricant sufficient for one year's operation. After every 2000 hours of operation, repack bearings with cup grease. Do not over lubricate. If power grease gun is used, avoid forcing grease into motor so that it comes in contact with armature. Check for worn parts.

2) Sleeve Bearing Motors—Fill oil cups before operating. Oil each 2000 operating hours (once or twice a year). Do not over lubricate. Sleeve bearing motors are wool-waste packed and need little oil. Check end play of shaft. If excessive, replace thrust washers.

C) Overload Protection

A wide swing in voltage or a change to lower or higher voltage may cause overheating and serious motor damage. Check plant voltage condition. A separate manual starter with thermal overload protection device for each unit is recommended.

Condenser

A) Cleaning

Clean condenser at least once a year; more often under unfavorable conditions. Unless condenser is kept reasonably free of dirt, lint and grease, its original heating capacity will be reduced . . . possibly to a serious degree.

Two commonly used cleaning methods are: (1) Loosen dirt by brushing fins on side where air enters condenser (side adjacent to fan on horizontal delivery units; outer side of vertical delivery units) and then turn on fan to blow dirt from unit. (2) Use high pressure air hose to loosen dirt by blowing *from side where air leaves condenser* (side adjacent to louvers on horizontal delivery units; side adjacent to fan on vertical delivery units).

For a more thorough cleaning of condenser, remove motor and spray a mild alkaline cleaning solution over the condenser. After a few minutes, follow by a hot water rinse. (A steam gun is used successfully for spraying cleaning solution and hot water.)

Condensers (especially steel) subjected to corrosive fumes should be checked and cleaned frequently.

B) Internal Corrosion Safeguards

1) Provide controlled water treatment—don't use excess of boiler compounds. (An excess of boiler compound carries over into steam lines; when such steam condenses, an acid is formed, which acid attacks parts of system nearest point of steam condensation, such as lower part of unit heater, also return branches and traps.)

2) Deaerate boiler feedwater. (Particularly if large amounts of new water are used.)

3) Insure rapid, continuous, and adequate condensate drainage by means of properly sized and installed traps and piping. Check traps for sticking. Clean strainers ahead of traps. (When traps don't work, condensate piles up in unit heater coil; a "quiet" water hammer results, and this breaks down coil tube joints.)

4) Adequately vent each unit.

5) Use low pressure steam when possible.

Casing

A) Cleaning

Periodic cleaning of casing is recommended to remove dirt, grease, and corrosive substances which might injure finish. Rusted or corroded spots should be cleaned and repainted.

Tighten fan guard and motor bracket. Check fan for proper clearance, free rotation, and firm connection to shaft.

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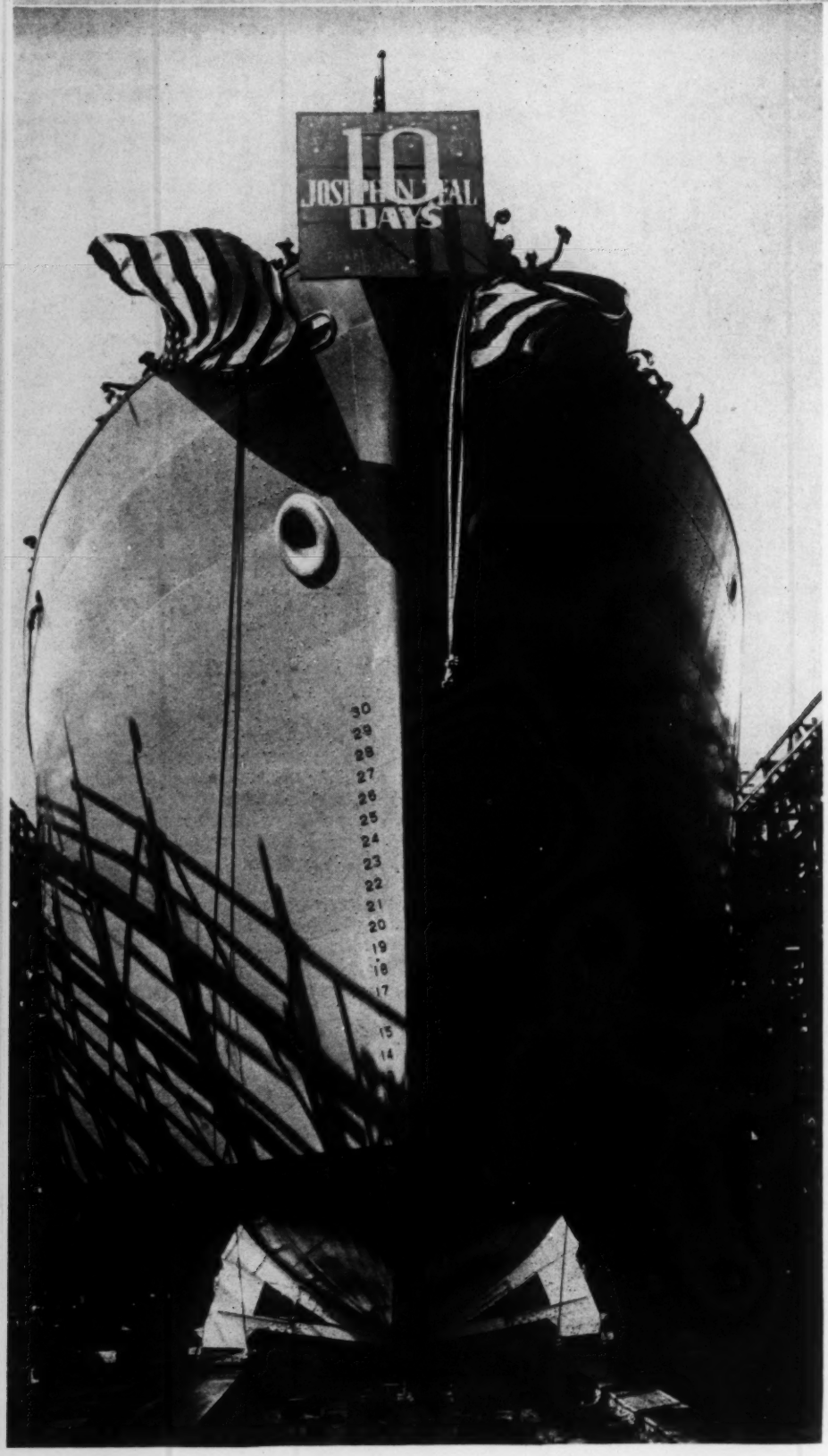
Heating the SHIPS FOR VICTORY



John Clarke, Marine Engineer, Explains Heating System Piping Requirements and Equipment Design for Our Merchant Fleet

●

SUMMARY—Cargo ship construction is one of the vital elements of our national war effort. To meet the demands of victory, all-out ship production is required, and many engineers, contractors, and suppliers are now engaged in marine design and construction whose previous experience has been largely with land installations. . . . Mr. Clarke—head of the heating, ventilating, and refrigeration unit of the engineering plan approval section, Great Lakes regional office of the U. S. Maritime Commission — explained in the March HPAC the requirements of marine heating installations, design temperatures and heating calculations, ship heating services and control, and gave information on proper submittal of plans for government work. . . . This month, he covers design and installation of the heating system piping and equipment design. A third article will take up hot water heating systems and piping insulation. . . . His material pertains primarily to cargo ships, and where Navy or Army work is concerned the designer should be particularly careful to familiarize himself with the special standards and requirements of the armed services

IN DESIGNING and installing the heating system piping for merchant ships, the following should be kept in mind:

1) The heating system is one of the last systems to be installed on the ship. Therefore, check as often as possible during the design to see that there are no interferences with other systems, equipment, the structure, and structural changes.

2) In running the piping, keep in the machinery spaces, working spaces, and passages as much as possible so that the minimum of piping will be in living spaces. To provide headroom it is desirable, and often necessary to follow the camber (crown of deck) and sheer of the ship. Because of this, it is usually best to feed the branch piping from mains run port and starboard wherever practicable. Where this is not possible, the next best thing is to run the piping outboard from port and starboard passageways. The supplying and draining of both the port and starboard sides of the ship by supply and return mains on one side should be avoided. Generally,

the best method on a cargo ship is to run the main up the machinery casing and then run branch mains port and starboard under each deck as necessary. It is not customary in marine work to make the runouts to the heating units larger than the risers. This is partly because the steam is usually dry, and also to simplify this installation.

3) The steam heating installation should ordinarily be divided into two systems: One for "constant service," which would include steam to the galley, domestic water heating tanks, steaming out services, and similar services for which steam must be available throughout the year; the second, for heating the working and living spaces. By dividing the piping in this way it is possible to cut out the quarters heating in the summer months without interfering with the "constant services." On a system of any size, and this will often include cargo ships, sectionalizing valves should be provided for the heating system branch mains so as to permit the repair of equipment in one section without affecting the entire system.

4) Steam and drain piping should be kept out of the cargo holds or any other inaccessible stores spaces where a leak would cause serious damage. In like manner, the piping must not be above or too near to switchboards or other electrical equipment, if practicable; but where it cannot be avoided, means should be provided to prevent water from dripping upon and injuring this equipment.

5) Watertight structure should be pierced as little as possible. This can be accomplished by leading mains to the various zones and taking off the branch mains and risers to that zone, within the zone. All deck and bulkhead connections should be kept to the minimum. Where necessary, the connections shown in the accompanying sketches may be used, except that the connection shown in sketch *d* may be used only for bulkheads and decks above the "bulkhead" (watertight) deck. An exception is made in the case of tugs in that the connection in sketch *d* may be used, as approved by the supervising authorities, through the main deck and through the bulkheads below the main deck.

6) To minimize the possibility of leaks, all joints should be kept to the practicable minimum with sufficient unions and breakable joints to permit ready installation and repair. In shipwork generally it is customary to bend all piping wherever possible. Fittings are resorted to only under duress; this has the advantage of providing lighter piping and reducing frictional losses.

7) In general the piping should be pitched as for shore work, keeping in mind the fact that the sheer and camber, as well as the trim of the ship, must be considered. Usually a ship will be trimmed by the stern, and it may be difficult to properly pitch a pipe against this trim. A boat rolls considerably, and in a storm may hold a list of as much as 15 deg where a quartering wind prevails. Consequently, level piping may sometimes be more desirable for transverse lines. Saturated steam lines should, of course, be pitched in the direction of flow, if practicable, but where superheated steam is available the drainage can be against the flow. When the steam is not flowing, the condensation will drain properly. Under the usual flow conditions the pipe will be dry, or reasonably so. Adequate drips should be provided in any case for all pockets, and at the base of the risers.

8) Expansion is a problem that should be given thorough consideration. But at the same time, the engineer must realize that he is not dealing with large pipes and concrete walls. On the usual cargo ship the supply main at the reducing valve will be 3 in. at the most. The branch mains will seldom be more than the 1¼ or 1½ in. size. Considering this, and the fact that it is usually impossible to run any length of pipe absolutely straight between bulkheads, the expansion problem more or less takes care of itself. Where straight runs do exist between two fixed connections, expansion bends or other approved means of expansion should be provided. Any special expansion connection should have the approval of the various regulatory bodies.

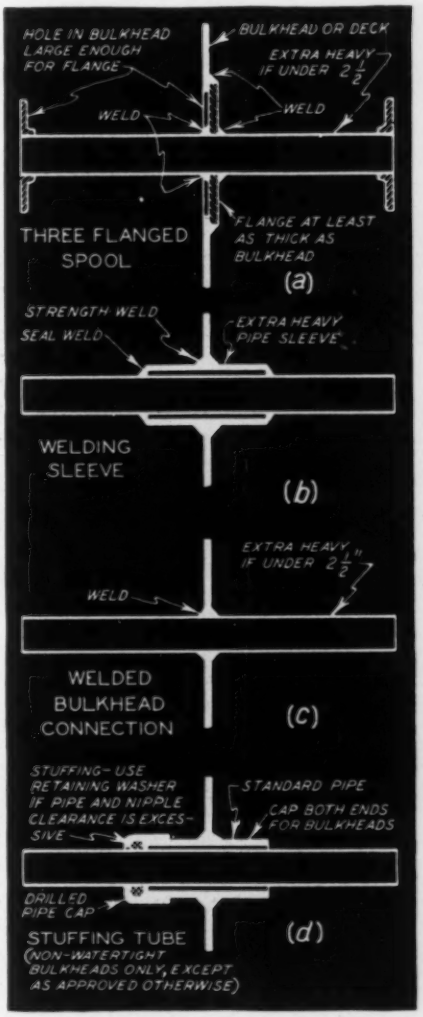
9) All steam units should be provided with a stop valve on the supply and a gate valve on the drain. The drain valve is preferred on radiators but is sometimes omitted. All units should also be provided with steam traps in the return ahead of the drain valve. Control valves should be

Schedule of Pipe, Joints, Valves, and Fittings

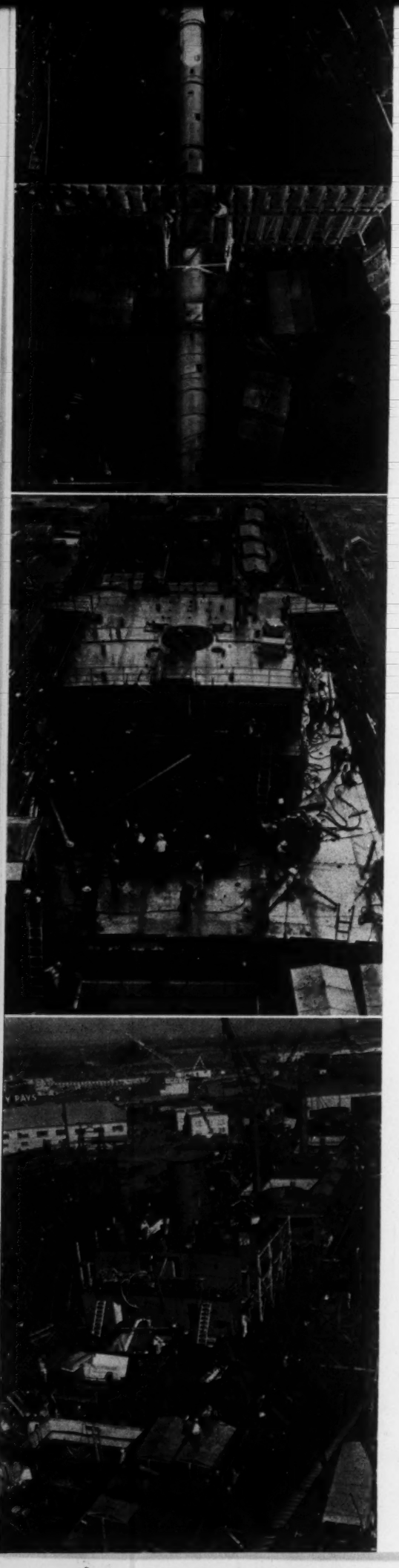
(from Schedule for Pipes, Joints, Valves, and Fittings, USMC Plan No. S48-26-1 dated May 19, 1939, Alt. 2)

PIPE	
Service	50 psi gage pressure maximum
Size	All sizes
Material	Seamless steel, lap welded, or electrically butt welded steel
Commercial standard pipe used	125 psi
Test Pressures—	
Shop (hydrostatic, for equipment)	Twice the working pressure
Installation	65 psi pressure with steam
JOINTS	
Size	2 in. and above — 1½ in. and below
Type	Flanged* — Ground union
Attachment to pipe	Socket type, welded; or screwed and welded — Screwed
JOINT MATERIAL	
Flange	Forged steel — Malleable iron
Nuts and bolts	Steel — Steel
Gaskets	Asbestos — Asbestos
VALVES	
Type	All — All
Connection	Flanged* — Screwed
Body	Cast Iron — Bronze regrinding
Mountings	Bronze — Bronze
FITTINGS	
Connection	Flanged* — Screwed
Material	Steel, cast — Malleable iron

*Exceptions made on individual merits of installation



Bulkhead and deck connections for heating system piping (does not necessarily apply to other ship piping systems). For 2 in. and above, flanged joints are required. Screwed flanges 2 in. and above must be seal welded. All connections shall be long enough so that couplings, flanges, unions, etc., will be accessible for repair without cutting insulation or sheathing on decks or bulkheads



Steam and Condensate Capacities for Heating System Piping

Standard weight pipe size, in.	Internal diameter, in.	Steam velocity, Fpm	Steam capacity, lb per min at given gage pressure						Condensate, lb per min
			15	20	25	30	35	50	
½	0.622	3830	0.58	0.67	0.76	0.85	0.95	1.21	2.99
¾	0.824	4420	1.18	1.37	1.55	1.74	1.92	2.45	6.03
1	1.049	4975	2.15	2.49	2.83	3.16	3.50	4.47	11.00
1¼	1.380	5720	4.28	4.95	5.62	6.28	6.95	8.89	21.9
1½	1.610	6170	6.28	7.27	8.25	9.22	10.2	13.0	32.2
2	2.067	6980	11.7	13.6	15.4	17.2	19.0	24.3	60.0
2½	2.469	7640	18.3	21.2	24.1	26.9	29.8	38.0	93.8
3	3.068	8530	31.5	36.5	41.4	46.3	51.2	65.4	161

Note: The above steam capacities are for essentially dry steam, as is usually supplied. For wet steam (from waste heat boilers or a contaminated evaporator) the above steam capacities should be reduced about 25 per cent. These capacities should not be used for heating systems wherein the condensate is returned to a heating boiler by gravity. Consult the ASHVE Guide for such design.

Allowable velocity of steam = 4860 $\sqrt{d}$ fpm. Allowable capacity of steam = 0.005455 Vd^2D lb per min. D = density of steam: lb per cu ft, and d = inside pipe diameter, in.

Allowable capacity of condensate (considered at 212 F) = 0.3263 Vd^2 lb per min, based on V (in fpm) = 30 $\sqrt{d}$ with the pipe considered flowing full (the low factor allows for the considerable flashing in the pipe).

preceded by a Y strainer and the supply stop valve. The steam traps (excepting radiator traps) should be preceded by 6 ft of unlagged pipe, and by a dirt pocket or Y strainer.

Materials and Connections

The accompanying table is taken from Maritime Commission plan No. S48-26-1, dated May 19, 1939, Alt. 2, Schedule for Pipes, Joints, Valves and Fittings. Several modifications have been made, however, to take care of changes which are not noted on the drawing. There are some exceptions in design, materials, and test pressures which are allowed, particularly for expediting present construction. The schedule lists preferred design and materials. Substitutes which the designer or contractor feels to be legitimate and justified (considering the operating condition as to service, pressure, vibration, and shock) should be proposed through the shipbuilding contractor for approval by the Maritime Commission office concerned. The engineer should also check to be sure that what he proposes is in accordance with the rules of the classification society (the American Bureau of Shipping) and the United States Coast Guard, Division of Marine Inspection and Navigation.

The attention of the engineer and designer is also called to the fact that globe valves with regrindable discs are used in marine work in lieu of replaceable composition discs. Exceptions have been permitted because of the present conditions, but regrindable discs should be provided if at all possible. All radiator valves are included in this category.

It is not recommended that standard pipe below ½ in. be used anywhere in the system.

Sizing the Piping

In sizing the steam and condensate piping it is recommended that the allowable capacities be taken from the table of steam and condensate capacities shown here. These capacities are specified by the Maritime Commission for exhaust and heating steam, and are in conformity with the usual marine practice. With dry steam, these capacities will give about a 5 psi pressure drop *from the source of steam to the heating units.* The capacities are given in pounds per minute to avoid large numbers, and to tie in with the rest of the power plant heat balance, which is given in pounds of steam per hour. It is recommended that all capacities be given in Btu or pounds of steam. Capacities in EDR should be avoided because this does not apply to other units and services on the system.

Some allowance should be made for wet steam in using this table, and a 25 per cent reduction in *steam* capacity is suggested. However, the full velocities seem to have been used on a number of ship installations without any known detrimental effect. Some designers who are familiar with low pressure steam systems may prefer to keep all branch mains a minimum of ¾ in. IPS. This will not be necessary nor desirable where the boiler water is deaerated and treated, and where the steam is dry. It is not recommended that the table be used for pressures under 15 psi gage unless the steam is superheated. For lower pressures and, in particular, where gravity drains to a heating boiler are used, the design recommendations of the ASHVE Guide should be used. However, the engineer should be careful to take full advantage of the static head available between the lowest heating unit and the boiler water level. In a land job this difference might be only 2 or 3 ft; in many marine installations it might easily be 10 or 15 ft.

Design of Equipment

The following summary will give the designer a good idea of the type of equipment desired for marine work:

Air Heaters

The air heaters should be of the copper fin and tube type with a header provided at each end of the

tube (excepting one type of antifreeze heater). The return bend type of heater must not be used. All preheaters (handling outside air) must be of the antifreeze type employing distributing tubes within the fin tube. One manufacturer makes these with dead end tubes, and header within a header. All preheaters must be installed with the tubes in the vertical position. The reheaters, preferably, should be installed vertically, if possible, but this is usually a difficult thing to do. The heaters should be sized on the basis of the steam pressure at the heating unit (8 to 15 psi lower than the main pressure at the source), and with a maximum allowable friction drop (per heater) of 0.18 in. of water. The face velocities for preheaters should be limited to 800 to 1000 fpm; and for the reheaters, from 650 to 800 fpm. Unless the heaters are provided with a bypass for summer operation, the *face velocity and allowable friction* through the heater should be based on the *full air capacity*. The *heating capacity* should be based on the *reduced volume provided during the heating season*. It may be desirable to allow about 10 per cent of capacity as a safety factor for reduced heating effect caused by corrosion and dirt.

Radiators

The radiators should generally be of the header type using copper finned tubing. If orifices are provided, care should be taken that they are properly sized for the operating conditions; better still, omit them. The radiator enclosure fronts should be No. 18 and the sides and top No. 20 USSG. One radiator which has found considerable favor is made from a copper pipe coil expanded by hydraulic pressure into steel finned cabinet sections. Cast iron radiators are not generally desirable because of their weight, bulk, lack of resistance to shock, and the fact that they are restricted to 15 psi gage operating pressure. The Liberty ships were first designed with cast iron radiation but it is believed that many of these ships are now being provided with copper convectors. In general, the radiators should be of the wall type, not more than 26 in. in cabinet height and 30 to 36 in. long. No radiator over 48 in. in length should be used. *In no case should pipe coil radiators be provided for the pilot house or other spaces.* Radiators within the circle for fixed objects for the standard compass must be made of nonmagnetic material.

Unit Heaters

The unit heaters for the engine room, steering gear room, and working spaces should be of the propeller fan, blow through type designed with copper fin tubes and a header top and bottom. It is important that good drainage be provided for these units as they might easily freeze up if shut off and water is trapped at the unit. Unit heaters are controlled manually.

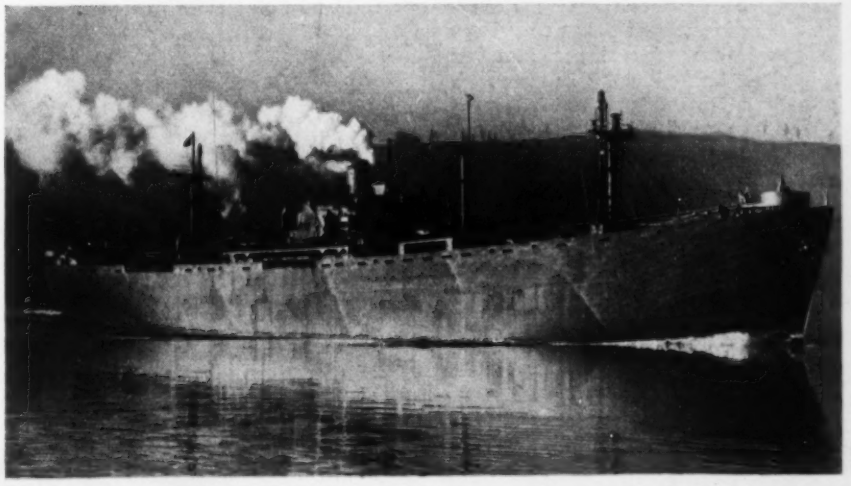
Traps

The traps usually provided are as follows: Inverted bucket, float-thermostatic, impulse, and thermostatic. The air heaters should be provided with either an inverted bucket, or float-thermostatic trap. No exception to this should ever be made for the preheaters, so as to minimize the danger of freezing. Since the reheaters are small, the aforementioned comparatively large traps may be awkward to install, and in this case the ordinary thermostatic, or an impulse, trap will give excellent service. In all cases, the trap selected should be compact and light and should be the smallest size available for the capacity and operating conditions. Generally the capacity requirements of domestic water heaters are high, and so thermostatic traps will not be suitable for the service. In sizing the traps, figure on an 8 to 15 psi drop in the supply piping to heating units with modulating thermostatic control valves, and 5 psi drop in the piping to units controlled by stop valves. For systems draining to an atmospheric tank the back pressure may be taken as 3 to 5 psi. Bypasses are not required around the traps because it is believed that they can be easily repaired, the parts replaced, or the trap mechanism held open for operation.

Control Valves

The control valves for domestic water and the ventilation air heaters may be of the self-contained, pneumatic, or electric type, except that the latter should be selected with care. At present there is only one manufacturer who builds an electric control which has been accepted for marine use by the Maritime Commission. Self-contained valves are advantageous in many cases because of their simplicity and are usually provided for the domestic water heaters unless the ventilation heating system is already controlled by electric or pneumatic valves, in which case a similar valve may be provided. The electric and pneumatic systems have the advantage of flexibility of design and may be of distinct advantage in some cases for this reason. The type of control for ventilation air heating was discussed in the March HPAC. Where domestic water heating is concerned, a modulating control is usually provided. However, where the heating steam is provided from a separate steam heating boiler, a *two position* control valve should be provided so as to eliminate unnecessary cycling of the boiler during summer operation. The thermostat should be set to open the control valve when the storage tank water temperature drops to 125 F, and close it when the water temperature rises to 160 to 180 F. The above is very important to keep in mind, particularly so where forced circulation, coil type heating units are concerned.

In sizing the control valves, the Commission will insist that oversizing be avoided. It is recommended that the designer or contractor consult the control valve manufacturer in this regard and accept his recommendation. The manufacturer should be provided with all pertinent data with any safety factors that may have been added clearly indicated or, better still, eliminated. It must be remembered that oversizing a valve with steam at 20 to 35 psi pressure will have a very detrimental effect on the life and operation of the valve. For the same reason, all control valves should be designed with a poppet or disc that is capable of metering the steam under the design conditions. Ordinary flat, or nearly flat, discs are not acceptable. It is also believed that single seated valves are the most desirable for the comparatively low capacities required by marine service. As an added precaution, keep in mind that the ventilation heating systems operate at reduced fan speed in the heating season, and the reduced volume should be used in calculating the steam capacity requirements. Do not use catalog ratings, which may represent the maximum and not the operating capacity.



Bypasses are not required nor installed around any of the control valves. So far as is known, all of these valves fail in the open position and thus it is still possible to operate the system with the stop valve. The repair or replacement of the damaged parts is not so difficult that the system couldn't be shut down for that length of time without any real harm being done. This would apply only to the controls of the heating system and would not include the reducing valves to the heating system, if such are provided, or any other control valves on the ship.

Electrical Equipment

All the electrical equipment should be suitable for marine use. A brief consideration of the constant marine problems of vibration, condensation, and the effect of corrosive salt atmosphere will make this readily apparent.

Electrical equipment should be tropically insulated and metallic parts should be noncorrosive, or covered with a noncorroding coating. Installations of the electrical equipment should be drip proof, moisture proof, or totally enclosed as applicable. Drip proof installations should be of such design that condensation cannot drop upon the electrical parts when they are tilted 15 deg constantly. The enclosing cases should permit drilling and tapping to take a screw type terminal tube or compression cable fitting. The use of knockouts is not permitted in marine practice because they soon rust out and leave the equipment unprotected.

In general, the motors should be of the stabilized shunt type unless otherwise specified or permitted. Control equipment should be positive acting so as to avoid sparking. The power available on most merchant ships is 120 and 240 volts d-c; however, this should be checked, inasmuch as some of the special type vessels are equipped for only 120 volts. Tankers may deviate from this in having 440 volt, 60 cycle a-c, but this is the exception rather than the rule in merchant practice.

In general, controls for motors should be provided with undervoltage protection in order that the motors will not restart on restoration of voltage. This is done in order to relieve the generators of a large starting load after breakdown. In general, controllers for motors are required to have separately fused control circuits. The control circuits should be fused on both sides of the control line. This is a deviation from the usual commercial practice. Whenever safety cutouts are necessary, the manual reset type should be provided unless it is specifically instructed to do otherwise.

Unit heater and similar small heating motors are not always required to be of the full marine type but in any case must be acceptable and approved for marine use.

It is recommended that those who are unfamiliar with the broad marine electrical requirements of merchant practice get a copy of the American Institute of Electrical Engineers standards No. 45. If any doubt still exists as to acceptance to the requirements, call in person or write the nearest Maritime Commission office. It cannot be over-emphasized that electrical equipment must be specifically designed to hold up under marine operating conditions.

Equipment, General

1) Care should be taken in sizing the heating units that they are sized for the operating conditions. With 35 psi steam pressure, unit heater capacity is increased over 25 per cent over the usual condition, air heater capacity (figuring 20 psi pressure at the heater) about 12 per cent, and radiator capacity around 70 per cent. Consequently, this represents an appreciable correction. The radiator and unit heater corrections are given in the ASHVE Guide. The corrections for air heaters, domestic water heaters, etc., should be taken from the manufacturer's data.

2) In sizing the various heating units (including control valves) and where the next size unit over the estimated required capacity may be considerably oversized, serious consideration should be given to using a unit (if available) a little under the estimated required capacity.

3) All steel parts of equipment must be well protected against corrosion. Generally it is required that ferrous parts be *hot dipped galvanized after fabrication.* Where this is not possible or where time does not permit, consideration is given to the various other rust resistant coatings commercially available. Where ferrous and nonferrous materials are in contact, it may be necessary in some cases to provide protection against electrolytic action, promoted by the salt atmosphere.

4) Spares must be provided approximately as follows:

a) Electrical spares in accordance with AIEE standard No. 45.

b) One valve, trap, strainer, etc., of each size and type; and 5 per cent of renewable parts such as control valve discs, seats, and power elements; strainer baskets; steam trap thermostatic elements, seats, etc., as required. Not less than one of each of these parts should be provided.

c) Sufficient spare parts, supplies, and equipment as necessary to provide for one year's operation. A complete set of V belts where such are provided.

[Photos courtesy Oregon Shipbuilding Corp.]

SAFE STORAGE OF COAL

(From a recent talk by Thomas J. Thomas, Associate Deputy Solid Fuels Co-ordinator for War)

"There are some technical problems connected with the safe storage of coal and full information on this subject can be secured without cost by application to the Bureau of Mines of the Department of the Interior, Washington, D. C. If well recognized practices are followed, you can store most coals in either your bins and bunkers or outside with safety and usually without much difficulty.

"Anthracite coal involves few storage problems. Bituminous or soft coal requires more care in the storage of large quantities in order to prevent the spontaneous generation of heat which, if allowed to rise to a burning temperature, might cause fire.

"The practice of certain elementary precautions will permit the safe storage of most bituminous coals. Some coals require greater care. If you are buying soft coal, your dealer should be able to furnish you with the information on its storage characteristics.

"While I do not propose to discuss at any length these soft coal storage problems, I will indicate their main outline and the suggestions for meeting them. The principal steps in securing safe soft coal storage lie in making sure that the coal is free of foreign matter and is stored in a satisfactory place and that it is piled in a compact mass so that air circulation through the pile is cut off as much as possible. The coal storage pile should be away from external sources of heat, such as steam pipes.

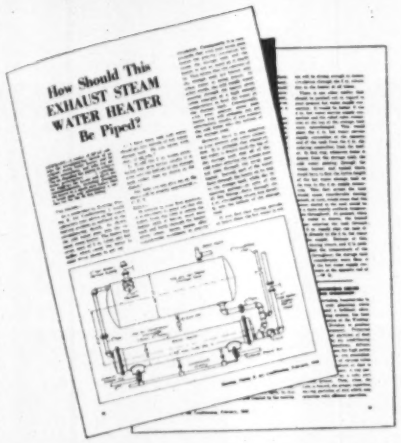
"Avoid all straw, paper, sticks, and oil rags in the coal and in the bins, and the ground, if it is stored outside, should be thoroughly cleared. In bins and bunkers old coal should be shoveled out of corners thoroughly before a fresh supply is introduced.

"Layering the coal avoids segregation of various sizes; store slack coal in layers of about 2 ft. If it is in a bin or bunker, it is best to have the container airtight on sides and bottom. A layer of fine coal dust over the top is additional protection for coals that tend to heat readily. The use of the chutes in filling bins and bunkers should be watched to make certain that there is no segregation of sizes and that the coal in each layer is thoroughly mixed.

"Good grade slack bituminous coal stored outdoors in small, low piles about 6 ft high ordinarily gives little trouble. If it is stored in higher piles, the coal should be laid in layers, avoiding segregation of sizes, and each layer should be compacted with a caterpillar, road roller, or similar equipment. The sides of the pile should be carefully compacted. A layer of fine coal dust, held down by nut sized coal, may be placed on top of the pile, or a sprayed layer of asphalt emulsion or similar material is effective."

"Open for Discussion"

HOW SHOULD THIS EXHAUST STEAM WATER HEATER BE PIPED ?



SUMMARY—A reader of HPAC ("C. F. T.") asked for advice on the piping connections for an exhaust steam water heater which was giving trouble. His questions and one reply were published in the February issue of Heating, Piping & Air Conditioning . . . Several other readers have been kind enough to prepare their recommendations for solving C. F. T.'s difficulty, and their answers to the problem are published here for the assistance of engineers and contractors who must make such installations work properly and who are often called upon these days to adapt the equipment at hand to the job.

THE QUESTIONS

As a subscriber to Heating, Piping & Air Conditioning I would appreciate your advice on the water heating problem as outlined in Fig. 1. As shown, the cold water goes into the exhaust steam heater. The heater has seven rows of 2 in. tubes and has baffles which cause the water to make seven passes to get out.

1) I have been told cold water should go into bottom of hot water storage tank. Do you agree with that; if so, why?

2) Would the 2 in. house circulating line give better results if it went into the bottom of the storage tank rather than as shown in the sketch?

Any help you can give me on the above will be greatly appreciated.—C. F. T.

USE SEMI-TANKLESS HOOKUP, SAYS E. L. MALM[1]

The information in the problem as submitted does not furnish very much data about the construction of the old air compressor intercooler which is being used for a water heater, but it seems reasonable to assume that it is a multiple pass, shell and tube unit with the water flowing in the tube and steam in the shell. If that is true, little could be done to change the unit from a multiple pass unit to a single or two pass unit, which would be more desirable, and in fact essential, to obtain proper circulation by gravity through the heater and tank.

It is my thought that a multiple pass unit could be made to operate effectively if hooked up to the storage tank, using what is called a semi-tankless hookup, as shown in Fig. 2. This figure is diagrammatic only—connections to the heater would conform to the available openings. I would suggest, however, that an opening be cut in the shell so that the steam inlet could be at the top of the shell and the condensation outlet on the bottom.

With the semi-tankless hookup, it is obvious that during standby periods when hot water would be stored up in the tank, circulation through the heater would be induced by the circulating pump controlled by a hot water control installed in the tank. Obviously, during periods of hot water draw in the building, the city water pressure would afford the motive power for overcoming the pressure drop through the heater.

E. A. HAHL[2] SUGGESTS A CIRCULATING PUMP

Neither of the two questions asked can be answered "yes" or "no." The combination of a tank and a water heater is used to accomplish a certain objective and the piping arrangement itself will vary depending upon what is to be accomplished.

There are two general schemes, which are shown as Figs. 3a and 3b. Fig. 3a shows the layout depending entirely upon recirculation between the heater and the tank. It is essentially the same as having the heating

[1]Bell & Gossett Co.

[2]American District Steam Co.

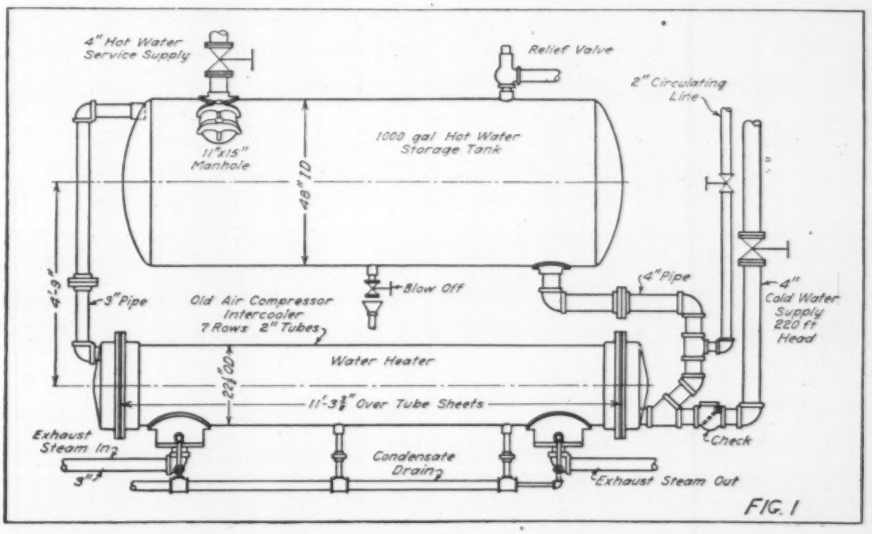


FIG. 1

element inside of the tank. Fig. 3b shows an arrangement in which all of the cold water first goes through the heater before entering the tank, and consequently the recirculating line is not intended to recirculate full capacity but merely to promote sufficient circulation to keep up the temperature of the tank in idle periods.

If Fig. 3a depends entirely upon gravity recirculation, the friction loss through the heater and the circulating piping must be very low or else forced circulation must be used. The Fig. 3b arrangement involves gravity circulation of only a small part of the rated load so that in terms of the full capacity the circulating system and the heater itself can involve greater amounts of friction drop than for Fig. 3a. Here also a small pump can be used for positive circulation in the recirculating line shown in Fig. 3b, if arrangements cannot be made that will permit suitable gravity recirculation.

It would seem, from the size of the tank and the heater, the fact that the heater has seven passes, and the fact that the circulating line to the tank is only 3 in., that the objective would have been in line with Fig. 3b—that is, the installation would be essen-

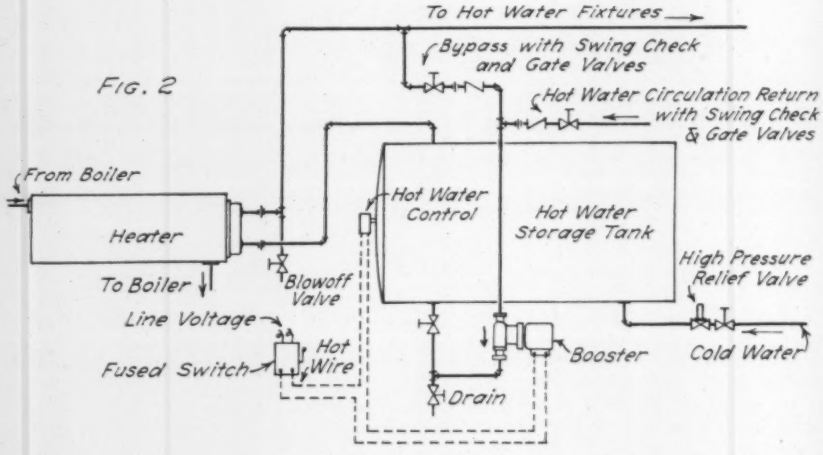


FIG. 2

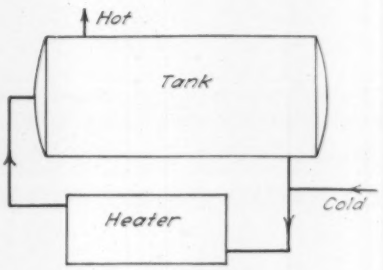


FIG. 3a

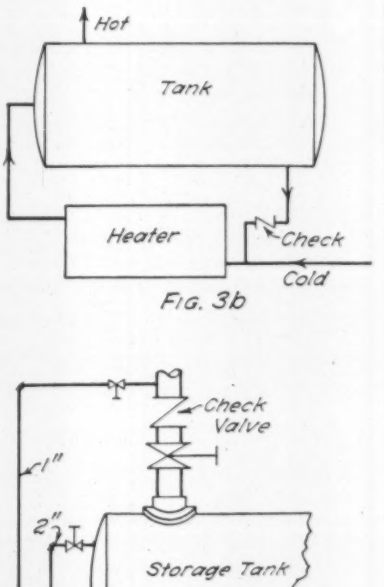


FIG. 3b

FIG. 4a

FIG. 4b

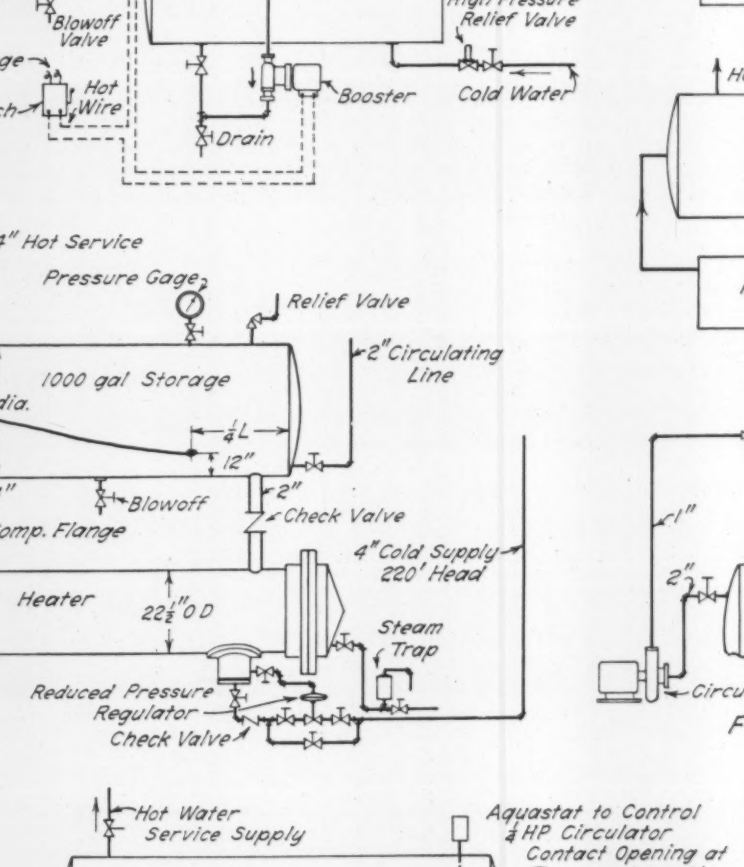


FIG. 5

tially an instantaneous heater with all of the water first going through the heater before it goes to the tank, and consequently the recirculating line is mostly a method of keeping the tank temperature up to the desired amount.

On the other hand, if the heater is altered to make it single pass, the arrangement indicated by Fig. 3a will be accomplished, but so far as obtaining results by gravity circulation are concerned, it would seem necessary to increase the size of the circulating piping because evidently there is very little chance to promote gravity recirculation under any circumstance, the tank being immediately above the heater itself.

I believe that the easiest way to get the most out of this heater is to put a circulating pump in the cold end of the recirculating line.

Regarding the question concerning the 2 in. house circulating line, "better results" might mean that better circulation is desired in the house circulating line or might refer to the possibility of better recirculation between the tank and the water heater. However, there is one very good reason why the house circulating line should not be led as shown in Fig. 1, which illustrates a purely gravity recirculating job, which depends for the movement of the water from the heater to the tank entirely on the difference in specific gravity between the water in the hot part of the recirculating system and that in the colder portion, which in this case is the 4 in. pipe. Therefore, if the water in the 3 in. pipe is as hot as possible and the water in the 4 in. return circulation pipe is as cold as possible, the maximum effect will be produced.

Assuming that the water coming back through the 2 in. house circulating line is warmer than the cold water supply, this cannot be introduced to advantage in the cold leg of the recirculating line. It would be better going into the tank itself. In Fig. 3a, which indicates a purely gravity recirculating arrangement, I have indicated the cold water coming into the system near the top of the recirculating return, because in that case (when no cold water is going into the system) it leaves this cold part of the circulating line in ideal condition to promote movement by gravity.

PAUL N. HUNT[3] RECOMMENDS CHANGES

The problem is not given in enough detail in that it lacks water usage, desired temperature of hot water out, temperature of cold water in, number and material of tubes in exchanger, and pressure of exhaust steam. It is assumed that both exchanger and storage tank, as well as all distribution lines, are adequately insulated. The 220 ft head seems high unless the system serves a tall building requiring this pressure. If not, the water head should be reduced by a pressure regulator allowing a minimum of pressure at the highest point in the building, thereby minimizing probability of accident, replacement of washers, etc.

In my opinion, the flow through the heater should be reversed, with water around the tubes and steam going through the tubes. This will allow for higher steam velocity and assure of no air binding of the heater, giving better heating. Second, pressure loss on the steam side is not important. Third, more heat will be recovered from the condensate as it will pass through the tubes leaving the heater at the coldest section of the heater, thereby increasing heat recovery from the condensate.

Referring to Fig. 4a, the hot water outlet should go directly to the bottom of the storage vessel as it will then hold the entire tank at a more even temperature, requiring less circulation of the storage tank to the heater and assuring of a more even temperature of hot water to the point of consumption. In order to control the temperature in the tank, which should be done, a thermostatic controller should be installed to control the inlet steam. The thermostat bulb should be installed approximately 12 in. above the bottom and one-quarter the length from the opposite end to the hot water inlet. With the thermostat control in service, it will not be essential that a steam trap be installed.

Cold water should enter the bottom of the heater at the opposite end from the hot water outlet, and should have a check valve in the line.

The 2 in. circulating line should enter the bottom of the storage tank. If connected to the cold inlet line, there is possibility of backflow of cold water through the 2 in. circulating line when hot water is being drawn from the line. For best results, a circulating pump should be installed, connected as indicated in Fig. 4b. This could be a very low head low capacity pump whose operating costs would be extremely small.

The 2 in. storage tank circulating line should be directly connected to the top of the heater with a check valve in the line to prevent bypassing of the exchanger by cold water.

The piping layout of Fig. 4a is much simpler than that shown in Fig. 1 and will give better control over the heating system.

In answer to the two questions:

1) The cold water should definitely go into the heater. If put into the bottom of the tank, short circuiting of the heater will result.

2) The 2 in. house circulating line should definitely go into the bottom of the tank as explained previously.

The entire problem could be simplified if coils were installed in the bottom of the tank instead of using the old heat exchanger for a heater.

[3]Mechanical engineer.

N. W. BLANCHARD SUGGESTS IMPROVEMENTS

I should like to present my views. First, there must be assumptions made, as follows:

1) Sufficient steam is available at suitable pressure to heat the required amount of water used from the tank even with seven passes in heater.

2) The 2 in. circulating line comes from a point below the farthest fixture or battery of fixtures, sufficient in height to cause recirculation.

As to C.F.T.'s first question, it would, in my opinion, be better to feed the cold water into the 4 in. return at a point of connection to the heater by use of a suction tee in lieu of the 45 deg fitting.

If a tee is not available a smaller size nipple can be fitted, by use of bushings, between the 45 deg fitting and check valve so that it projects into the 45 deg fitting and replaces the 4 in. nipple. One end of this small nipple would have a long thread on it of necessity. The amount of projection necessary into the 45 deg fitting would be such as would insure the cold water entering the heater and not the 4 in. return pipe. Should it be found that cold water does enter the 4 in. return, a check valve in the 4 in. return line would be of benefit. However, the aspirating effect should provide sufficient induced flow to pass both return and cold makeup water through the heater. In this installation I would prescribe introducing the cold water into the heater rather than into the storage tank so as to take advantage of the 220 ft head in providing circulation for heating when water is drawn from tank.

My first assumption will provide sufficient heating effect properly to heat the water as it passes through the heater. No temperatures of hot, circulating, or cold water are given, hence only reference to this assumption can be made in the analysis of piping connections. The suction tee effect could also be obtained through pipe welding if the threading and bushings are not desired. Any number of methods for manufacture could be devised.

It should also be noted that the 4 in. cold water supply at 220 ft head has a maximum water flow capacity and the suction tee will reduce this capacity at given head. Some gain in heating might be obtained if the size of the cold water inlet were reduced to allow passage of only the maximum water demand from the tank.

As to C.F.T.'s second question, I would suggest connecting the 2 in. circulating line into the bottom or side of the storage tank and not into the 4 in. return or near the cold water supply inlet. Again I refer to my assumptions, and presume that good recirculation is present when no water is being drawn from tank. Reason

for this connection is to prevent the 2 in. circulating line becoming a flow line for cold water when water is drawn from tank and to secure an intermediate temperature mixture, when circulating water enters tank, over and above the temperature of mixed circulating water and cold water. In other words, introduce the circulating water into the tank at its returning temperature without the necessity of reheating it through the heater. At periods of small draw from the tank, heat stored in the circulating water would become an asset in maintaining more uniform temperature in the tank. With no water being drawn from the tank the existing setup due to circulation between heater, 3 in. flow pipe, tank, 4 in. return pipe and back to the heater causes the water in the 2 in. circulating line to again pass through heater. This is not objectionable but a different procedure may result when water is being drawn from tank and it may mean the difference between good and bad operation.

It is noted that W.G. in his reply in the February HPAC advises the interchange of the 4 in. hot water service supply and the relief valve connections. It is further noted from Fig. 1 that the relief valve tapping is much smaller than 4 in., hence a new opening would be required at the relief valve location. If a new opening were to be made I would suggest that it be made at the top center of the tank and the 4 in. hot water service supply taken off from this new point. Whether or not a change here is necessary depends upon whether or not present water used is hot enough or too hot. It would of course be advisable to have the water in tank at somewhere near uniform temperature and this result can be secured by mixing within the tank; that is, heated water in and circulation water in. Any introduction of water colder than the recirculated water from the 2 in. line, such as cold water supply backing up, will give rise to various temperatures within the tank and more heating required by circulation through heater.

It is noted that the suction tee arrangement only functions when water is being drawn from tank, hence no induced circulation is available during periods of no draw. Heating is by gravity circulation only then and the revised piping above suggested gives two main circulating circuits as follows: One circuit through 4 in. hot water service supply to point of connection in system where 2 in. circulating lines take off, thence back to tank. One circuit through 3 in. heater flow pipe to tank, thence back through 4 in. return pipe to heater. This arrangement seems better to me than the original combined return layout and the changes necessary would involve little cost. No matter what connections are made it is possible to trace a third combined circuit also.

For a proper analysis, all water temperatures present and required, together with heating surfaces available, amount of water required per hour, pressure and amount of steam available, etc., should be considered.

Should the return water from the tank pass through the 4 in. pipe to the heater in a very hot state, some noise and water hammer, possibly excessive corrosion, may result due to the mixture of hot and cold water at the suction tee. The remedy would be to maintain the water in the tank at a lower temperature. This condition will only happen at very hot temperatures.

CARL BACHMANN[4] OFFERS HELP

I read the predicament of C.F.T. and the suggestions by W.G. in the February HPAC with great interest; however, these suggestions with the subsequent service shutdowns do not effectively accomplish the result sought by C.F.T. it seems to me.

In Fig. 5, I should like to be of assistance to convey to C.F.T. our remedy of an identical condition, which after the alteration provided ample hot water. In conclusion, I should like to add that if C.F.T. will utilize his ingenuity, he can perform the alteration with very little service interruption. In the event I personally can be of help, of course without any remuneration or obligation, please do not hesitate to call on me for exchanging our experience with other HPAC readers.

[4]American Hotel Corp.

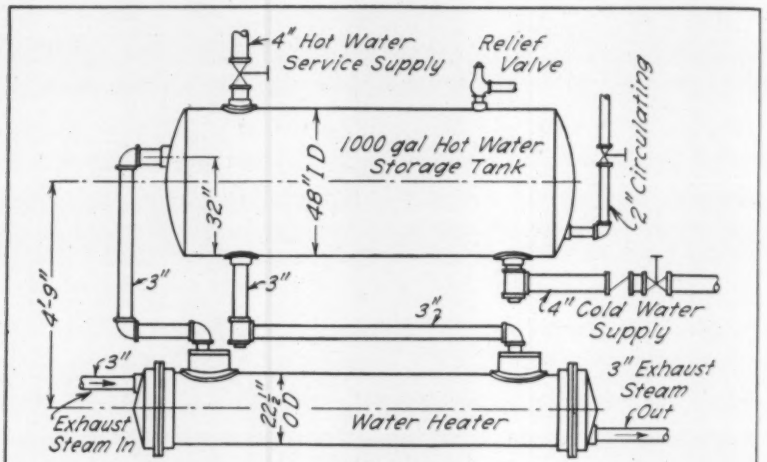


FIG. 6

HUGO C. PETERS[5] SUBMITS SUGGESTION

Fig. 6 is a sketch of a revised hookup for the storage tank and heater about which C.F.T. asked in the February HPAC.

I trust you will find this suggested piping layout of some merit.

[5]Heating engineer.

AUXILIARY OIL RATIONS FOR CARE OF THE SICK

The procedure under which those who require additional heating oil for the care of ill or infirm persons may obtain extra supplies was simplified last month by the Office of Price Administration. Under the plan, local war price and rationing boards may grant the additional heating oil rations to householders living in private residences and multiple dwellings heated only by oil stoves without requiring a certificate from a qualified plumber, architect, or heating engineer showing the amount of additional heating oil needed. The exception applies only if unreasonable difficulty in obtaining the certificate would be encountered. If the expert's certificate may be obtained easily, it still is required.

Where the certification is dispensed with in these cases, the local board will issue auxiliary rations equal to 6 per cent of the applicant's total yearly ration for each of the five thermal (heating) periods during which the extra oil is required. For example, if an allowable allotment for the heating year is 1500 gal, the board may issue coupons equal to 6 per cent of this amount, or 90 gal. for each thermal period that the consumer is eligible for the auxiliary supply. OPA officials estimated that the 6 per cent extra allotment will permit a temperature about 10 deg in excess of the temperature the normal ration would permit.

In all instances, the application for an auxiliary ration must be filed on the form designed for this purpose (OPA-1104). The amendment (No. 43 to ration order 11) became effective March 11.

How an Air Conditioned Office Building Reduced Its Heating Steam Consumption

W. C. Heimbeck, Building Manager, and M. Ehrlich, Building Engineer, Describe the Changes Made

SUMMARY—Conservation of heating steam is always good business — in wartime, it is a very practical and patriotic essential. While there are certain general principles of efficient heating, the achievement of economy in any building depends upon careful analysis of the peculiarities of the particular installation . . . The authors tell here some of the steps taken to reduce the steam for heating the International Harvester building from 0.93 to 0.43 lb per 1000 cu ft per degree day

WHEN THE International Harvester Co. purchased its headquarters office building in Chicago a few years ago, among the work done was the installation of an air conditioning system to serve the 15 floors which it contemplated occupying, consisting of 140,000 sq ft of floor space.

Thirteen thousand, five hundred cfm of air is required per floor, and is furnished by two units, each comprising two delivery fans, four rows of coils for summer cooling, and two rows of coils for winter heating. As the same piping serves both summer and winter water circulating, the changeover is accomplished by means of a three way air operated valve, controlled from a central point in the engineer's office.

The principal equipment, situated in the basement, consists of a 400 hp centrifugal compressor, a 1000 gpm chilled water circulating pump, two 500 gpm hot water circulating pumps, a 500 gpm condenser pump, an 80 gpm pump and heat exchanger for humidification, and a 1000 gpm heat exchanger for heating purposes. Controls are air operated, and their functions are directed from the engineer's office in the basement. To record the results of operation, flow meters and recording thermometers are provided in the engineer's office.

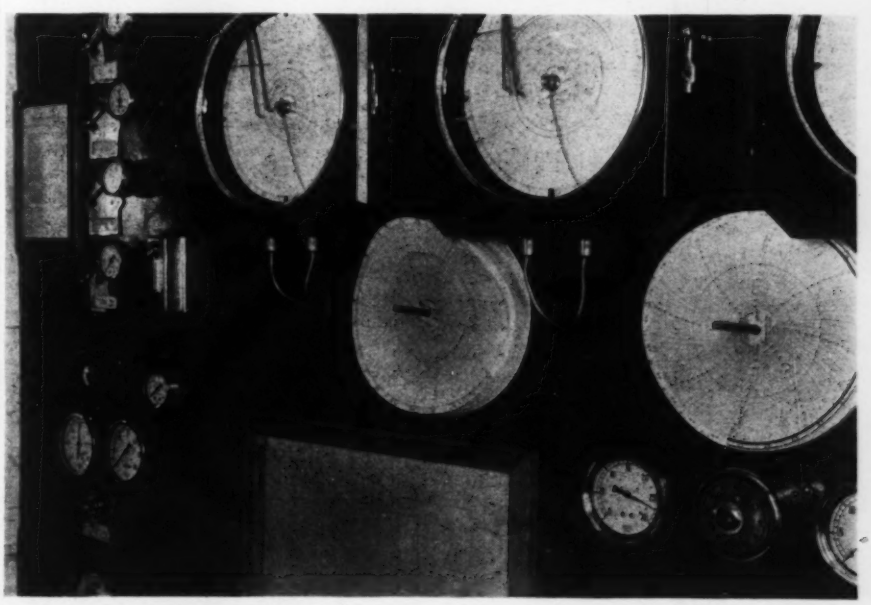
As installed in 1938, the system was intended to be used only for ventilating during the winter season, delivering air at a fixed temperature, using the existing steam radiators under the windows for heating the space. This resulted in overheating in mild outdoor temperature and underheating under severe conditions. To remedy this condition, a thermostat was placed in the return and outdoor air mixing chamber to operate the maximum outdoor air damper, fixing it at a minimum at 50 F, and converting the rest of the system to a full blast heating system, varying the temperature of the delivered air as the space would require. This proved satisfactory under a certain condition only, but as outdoor conditions changed, other problems developed, not the least of which was the fact that our boiler capacity was taxed to the limit to furnish steam to this equipment.

Our steam consumption for heating the building on the degree day basis, increased about 50 per cent. At this point, we made a careful study of all conditions in the building. We found that we had an internal heat gain (in addition to the radiators) of 160,000 Btu as against a heat loss of 180,000 Btu at zero outdoor temperature. To utilize this heat became our problem. Each floor is equipped with 52 radiators, having a total of 1200 sq ft of heating surface. From conditions prevailing adjacent to these radiators, it was indicated that too much heat was being emitted. To correct this condition, the valve stops were fixed so that the radiator valves would be limited in their opening, thereby cutting down the steam flow. The effect is that at no time are the radiators fully heated.

The next step was to have the maximum damper thermostat piloted by the room or return air thermostat so that the maximum outdoor air damper would not open unless the room temperature was in danger of rising too high, and in addition, arrangements were made to vary the temperature of the water circulating through the coils, with the controls in the engineer's office. This change had a very marked effect on the steam consumption, but did not account for all of the internal heat. Further investigation found that due to wind direction, the minimum outdoor air damper would supply 100 per cent outdoor air, and further, due to improper outdoor and return air mixing, we had a false reading at the suction side of the fans, varying about 70 F at zero outdoor temperature. To correct these conditions, baffles were introduced in the mixing chamber to obtain a better mixture of outdoor and return air, and to overcome the effect of wind direction, a thermostat was placed between the electro-pneumatic switch and the minimum outdoor air damper motor, setting a low limit on the mixing chamber temperature of 55 F. Fifty-five degrees was selected because about 25 per cent outdoor air at 0 F mixed with about 75 per cent return air at 80 F gave a mixture at approximately 55 F.

These changes resulted in reducing the steam consumption from 0.93 lb per 1000 cu ft per degree day to 0.43 lb. The building volume is 3,300,000 cu ft, total floor area is 225,000 sq ft, and height is 23 stories.

An instruction sheet and a control diagram are mounted on the instrument and control panel in the engineer's office



CONVERTING

Herman E. Winkler Gives a Quick Picture of

SUMMARY—Prompted by an apparent misapprehension in many quarters as to difficulties involved in engineering conversions from oil to coal, Mr. Winkler, executive vice-president of the U. S. Machine Corp., discussed the subject at a recent series of sales engineering meetings . . . The following is a part of his presentation

THIS ARTICLE is prompted by an apparent misapprehension in many quarters as to the difficulties involved in engineering a conversion from oil to coal firing. Let me say now that the procedure is neither mysterious, difficult, nor complicated. There are a few things which must be known and understood, but anyone willing to pay a small price in study can quickly acquire the necessary facts.

The starting point in any discussion about either oil or coal firing is the boiler. Basically, a boiler is just a heat catcher and transferer. It catches the heat released in the firebox and transfers it to the water. It doesn't matter whether the source of heat is oil or coal, so in this fundamental aspect there is no difference between oil or coal firing.

Heat Release and Efficiency

Heat release in the firebox of a boiler is a function of two factors, the weight of fuel burned, and the efficiency of burning. In combustion work there are two kinds of efficiency to be considered: burning efficiency, and overall plant efficiency. Burning efficiency has to do with what takes place in the firebox only and has nothing to do with the boiler outside the firebox. Overall plant efficiency combines burning efficiency with the ability of the boiler to catch the heat released in the firebox.

Components of Fuel

Burning efficiency is a function of the fuel employed, so let's see how oil and coal compare in this respect. Here is a sample "as fired" analysis of coal:

Fixed carbon	50 per cent
Volatile matter	38 per cent
Ash	9 per cent
Moisture	3 per cent
Sulphur	1 per cent
Heat content	14,000 Btu per lb

Here is a sample analysis of a particular fuel oil:

Carbon	83 per cent
Hydrogen	14 per cent
Sulphur	1 per cent
Nitrogen	2 per cent
Heat content	19,000 Btu per lb

The components which provide the heat in coal are the fixed carbon and the volatile matter. In oil, they are the carbon and hydrogen.

How Fuel Burns

Let us consider what happens when anything burns. Burning is the rapid combination of a combustible with oxygen, and takes place only at temperatures above the ignition point of the combustible. The higher the temperature, the more rapid the burning or oxidation. The result of the chemical reaction of oxidation is the liberation of heat and the formation of the oxide of the combustible. For example, the oxide of carbon is carbon dioxide, CO_2, and the oxide of hydrogen is H_2O, or water (actually steam at the temperature at which the reaction takes place).

Considering the combustion mechanics of coal, carbon is the principal heat generating ingredient. The oxygen required for combustion is supplied by the air sent into the firebox. By volume, air is composed, principally, of 21 per cent oxygen and 79 per cent nitrogen; by weight, the respective figures are 23 and 77.

When the oxygen in air combines with just the right amount of carbon, this is what happens:

C (carbon)	+	O_2 (oxygen)	=	CO_2 (carbon dioxide)
1 lb		2 2/3 lb		3 2/3 lb

From the above burning or oxidation process, 14,600 Btu of heat is released. (This is the theoretical value of complete combustion and is not possible in actual practice. We strive only to approach it.)

The above equation pre-supposes that the theoretically correct amount of air has been supplied to the combustion process. Let's see what happens if too little air, and therefore too little oxygen, is furnished.

Carbon has the propensity to accept partial oxidation. In this case, the product of combustion is carbon *monoxide*, or CO, instead of being carbon *dioxide*, CO_2, as would occur if oxidation were complete. The reaction then looks like this:

2 C (carbon)	+	O_2 (oxygen)	=	2 CO (carbon monoxide)
1 lb		1 1/3 lb		2 1/3 lb

In the above example, only 4400 Btu is released—a bad case of incomplete combustion.

Thus, at any time that conditions in the firebox permit the formation and escape of CO, the fuel has yielded only a part of the heat which it might have, had the carbon content been completely oxidized to CO_2. The presence of CO in flue gas, therefore, always indicates a loss in burning efficiency.

Excess Air Loss

On the other hand, too much air in the firebox also causes a loss in burning efficiency. You will recall that air is 79 per cent nitrogen, and nitrogen is an inert element as far as combustion is concerned. It will not oxidize (burn) in any temperature we can get in a firebox. Nitrogen, however, absorbs heat from the fire until it is the same temperature as the flue gas. It then goes the way of all flue gas—up the chimney and gone forever—carrying with it a large proportion of unused heat.

The same is true of oxygen, if more than necessary is furnished. Excess quantities can't find any carbon to combine with and accordingly move right out the stack loaded with heat.

Nothing can be done about this condition, except to exclude any more air than is necessary. Excess air moves through the firebox in an unchanged state, being raised in temperature only. It obviously reduces the percentage of CO_2 found in the flue gas and therefore lowers burning efficiency. Some excess air, however, is required in practice, for reasons explained below.

All the chemistry of combustion covered so far applies fundamentally to either coal or oil. If you realize how much alike the basic problems are, you will more easily recognize certain differences in the application of oil and coal burning equipment—differences which are necessarily involved in the conversion of one to the other.

It has already been indicated that an analysis of the flue gases would show the degree of burning efficiency. While there may be oxygen (O_2), carbon monoxide (CO), in addition to the nitrogen (N_2) in stack gases, it is the percentage of carbon dioxide (CO_2) which really counts.

Since there is 21 per cent oxygen in air, theoretically perfect combustion of carbon would put 21 per cent of CO_2 in the stack gases. But with coal as fuel, at least 50 per cent excess air is necessary in practice; therefore, a range of 12 to 14 per cent CO_2 is considered excellent.

Stack Gas Temperature

At this point, let us see what the difference is between the burning effi-

G TO COAL?

of Stoker Application to Oil Conversion

ciency and overall plant efficiency. If the firebox we have just discussed in connection with flue gas analysis and burning efficiency were out in a yard with no boiler attached, all the remarks about burning efficiency would still hold good.

But so far, no mention of stack gas temperature has been made. It is this temperature which largely affects the overall plant efficiency figure. Remember that the boiler is a heat catcher. If it does a good job of pulling down the temperature of the gases leaving the firebox, it is said to have a high absorption efficiency. If it does not do a good job of pulling down the gas temperature, it has a low absorption efficiency.

So fix these facts in your mind and you have the complete efficiency story:

1) The flue gas analysis, or CO_2 reading, indicates the burning efficiency of the fire in the firebox.

2) The temperature of the gases leaving the boiler indicates the absorption efficiency of the boiler.

3) A combination of the burning efficiency (CO_2 in per cent) and of the stack temperature (F) indicates the overall plant efficiency.

In other words, if gases leaving the boiler have a high CO_2 content and a low temperature, then the overall plant efficiency is excellent.

Differences Between Coal and Oil

Thus far we have been talking about conditions which are practically parallel in the combustion of coal and oil fuel. Let us now consider the points of difference between the fuels, as they relate to the conversion of oil fired to coal fired plants.

The physical states of the fuels differ. Oil is a liquid and coal is a solid. This difference calls for consideration of storage space when conversion to coal is contemplated.

The heat contents of the fuels differ. This difference in heat content must be considered in order to specify correctly stoker capacities for installations previously burning oil.

Coal has a mineral residue—ash—while oil does not. This factor obviously influences firebox design, as there must be a place for the ash to accumulate and firebox doors must be provided for the removal of fused ash or clinkers.

With oil firing, extreme firebox temperatures are desirable, even necessary, to obtain complete combustion. This condition is undesirable with a stoker because of the difficulty in handling ash in the molten or fused state. It is difficult to draw a parallel between the two fuels, as affected by firebox temperatures, except to say that with oil it is advantageous to have the entire firebox extremely hot, while with coal it is best to localize, as much as possible, the extreme temperature to the upper part of the fire and to the combustion space which is above the fire bed. Thus, a condition favorable to the solidification of ash into easily handled clinker is approached.

The fact that coal contains ash is the reason behind the major difference in details pertinent to the conversion from oil to coal firing. The presence of ash demands a firebox design for stokers which is not consistent with oil burner practice.

Quick Estimate for Conversion Job

In figuring a conversion job the difference in heat content of oil and coal can be adjusted by a simple calculation:

19,000 (Btu per lb) × 7 (lb per gal) = 133,000 Btu per gal.

Taking 13,300 Btu as the heat content for a particular coal:

13,300 (Btu per lb) × 2000 (lb per ton) = 26,600,000 Btu per ton.

$$\frac{\text{26,600,000 (Btu per ton of coal)}}{\text{133,000 (Btu per gal of oil)}} = 200$$

The heat content of one ton of such coal is thus equal to that of 200 gal of such oil. By substituting the correct heat content values for the particular coal and oil involved, the proper conversion factor for any installation will be obtained.

This conversion factor can be used in roughly sizing a stoker to replace an oil burner of a given capacity. It is important to remember that in sizing a stoker, that is, in selecting capacity for a given load, do not figure the stoker to run continuously. Figure the stoker size for a maximum operating time of 70 per cent.

Here is a typical example of selecting a stoker to replace an oil burner using 10,000 gal per yr:

10,000 (gal per yr of oil) ÷ 200 (factor for converting gallons of oil to tons of coal) = 50 tons per yr of coal.

50 (tons per yr) × 2 (factor for converting to lb per hr) = 100 lb per hr.

[Note: The factor 2 is arrived at by assuming 1000 hr operation per year]

Therefore, a stoker with a capacity of 100 lb per hr should be selected to replace the oil burner.

The above quick method is a bit too brief for good stoker engineering practice. It does not take into consideration that the 10,000 gal of oil previously used may be too much, because of inefficient operation, or too little to furnish a satisfactory amount of heat. The accurate method is to start from scratch, using the installed radiation as a guide to the size of stoker required.

Assume that 3000 sq ft of steam radiation is installed. (One sq ft of steam radiation emits 240 Btu per hr.) The calculation is as follows:

3000 (sq ft radiation) × 1.33 (factor for piping loss and boiler pickup) = 3990 sq ft EDR (equivalent direct radiation).

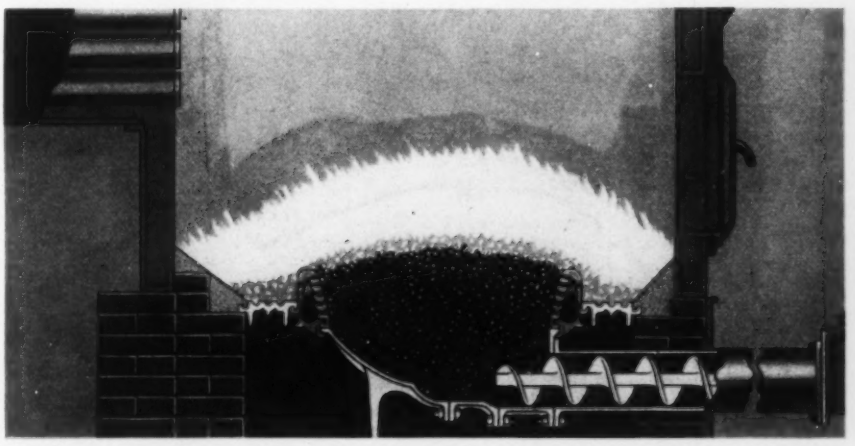
3990 (EDR) × 240 (Btu per hr per sq ft) = 957,600 total Btu per hr required.

13,300 (Btu per lb heat content of coal) × 0.65 (overall plant efficiency factor) × 0.70 (stoker operating time) = 6051 available Btu per lb of coal.

$$\frac{\text{957,600 (total Btu per hr required)}}{\text{6051 (available Btu per lb of coal)}} = 158 \text{ lb per hr of coal, required capacity of stoker.}$$

It is obvious from the foregoing that between oil and coal there is little basic difference in the combustion problem. The procedure of a heating survey or determining building heat loss is standard for both fuels. And the calculations required to size a stoker properly for conversion purposes are simple.

Sectional drawing of underfeed stoker



CALCULATION OF PIPE BENDS

By Peter J. Mascaro*

IN THE engineering data sections of various piping catalogs and handbooks there are developed formulas for the calculation of pipe bends. I believe such formulas can be simplified with a great saving of time for the drafting room; as an engineer, I have found that too much time is often spent by draftsmen in figuring pipe bends.

I have worked out two formulas for the general case of an offset bend, which can be used also in all other cases, as is shown in Fig. 1. These two formulas have proved very successful in our drafting room.

Fig. 2 shows the derivation of the formulas, and in Fig. 3 an example is worked out.

All other cases of pipe bends—such as angle bends, offset bends with unequal radii, etc.—generally developed as separate cases in the piping data books, can be solved applying the two formulas of Fig. 1. In addition, the new formulas also give a more exact value of the bending angle because there is only one approximate value which appears.

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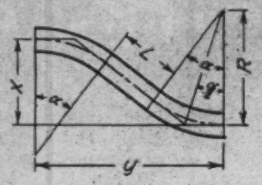
FIG. 1 – CALCULATION OF PIPE BENDS

(1)-OFFSET BEND-EQUAL RADII, GIVEN R, x, y



The following formulas are general and can be used in any case: R>x, R=x, R<x

$$(a)\ L=\sqrt{y^2+x^2-4Rx} \qquad (b)\ \tan\frac{\alpha}{2}=\frac{x}{y+L}$$

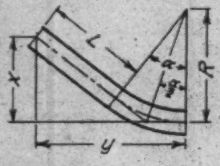
NOTE: If y is not given its value can be assumed, but either given or assumed its value must be $\geq\sqrt{4Rx-x^2}$.

(2)-EQUAL RADII, L=0

In this special case above formulas can be simplified as follows:

$$\tan\frac{\alpha}{2}=\frac{x}{y} \quad \text{and} \quad y=\sqrt{4Rx-x^2}$$

(3)-ANGLE BEND, GIVEN R, x, y.

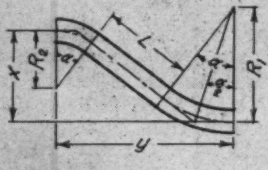


The Angle Bend can be considered as half of an Offset Bend and the values of L and $\frac{\alpha}{2}$ can be obtained directly from formulas (a) and (b) which in this special case change as follows:

$$L=\sqrt{y^2+x^2-2Rx} \quad \text{and} \quad \tan\frac{\alpha}{2}=\frac{x}{y+L}$$

where x, y, and L are the values for the Angle Bend as shown in the figure.

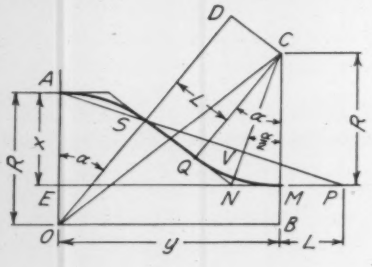
(4)-OFFSET BEND-UNEQUAL RADII, GIVEN R_1, R_2, x, y



Formulas (a) and (b) can be used putting in place of R the mean value of the two radii: $\frac{R_1+R_2}{2}$. The formulas change as follows:

$$L=\sqrt{y^2+x^2-4x\left(\frac{R_1+R_2}{2}\right)} \quad \text{and} \quad \tan\frac{\alpha}{2}=\frac{x}{y+L}$$

FIG. 2 – DERIVATION OF FORMULAS



Let AM be an offset bend of equal radii R, x be the offset and y the distance between the tangent points A and M, L the straight part of the bend and α the angle.

We want to find the relations between those quantities and in particular find L and α when the other quantities are given, which is the most common case.

In the right triangle OBC, BC=2R-x and the hypothenuse

$$\overline{OC}^2=\overline{OB}^2+\overline{BC}^2=y^2+(2R-x)^2 \qquad (1)$$

In the right triangle ODC, OD=2R and $\overline{OC}^2=(2R)^2+\overline{DC}^2=4R^2+L^2$ (2)

Substituting this value in (1)

$$4R^2+L^2=y^2+(2R-x)^2 \quad \text{or} \quad 4R^2+L^2=y^2+4R^2-4Rx+x^2$$

$$L^2=y^2+x^2-4Rx \quad \text{or} \quad L=\sqrt{y^2+x^2-4Rx}$$

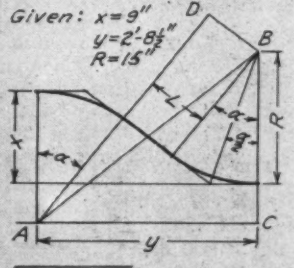
Draw the chord AS and continue it until it meets the line EM in P. The two right triangles NVP and NVS are equal, having one side NV in common and the two adjacent angles equal. Then NP=NS and deducting the two equal parts, NM and NQ, the part MP=SQ=L.

In the right triangle AEP the angle APE is equal to angle NCM because the two sides of the former are perpendicular to the two sides of the latter. But angle NCM=$\frac{\alpha}{2}$, then APE=$\frac{\alpha}{2}$ and

$$\tan\frac{\alpha}{2}=\frac{AE}{EP}=\frac{x}{EM+MP}=\frac{x}{y+L}$$

The above relation gives the angle $\frac{\alpha}{2}$ and multiplying by 2 we obtain the angle α.

FIG. 3 – EXAMPLE OF CALCULATION OF PIPE BENDS



$$L=\sqrt{y^2+x^2-4Rx};$$

$y^2=(2'-8\frac{1}{2}'')^2=$ 7.3351

$x^2=(9'')^2=$ 0.5625

7.8976

$4Rx=4(1'-3'')\times 9''=$

$4\times 1.25\times 0.75=$ 3.7500

$L^2=4.1476$

$L=\sqrt{4.1476}=2'-0\frac{7}{16}''$

$$\tan\frac{\alpha}{2}=\frac{x}{y+L}=\frac{9''}{(2'-8\frac{1}{2}'')+(2'-0\frac{7}{16}'')}=\frac{9''}{4'-8\frac{15}{16}''}$$

$$=\frac{9}{56.937}=0.1581$$

$\frac{\alpha}{2}=8°59' \quad \alpha=17°58'$

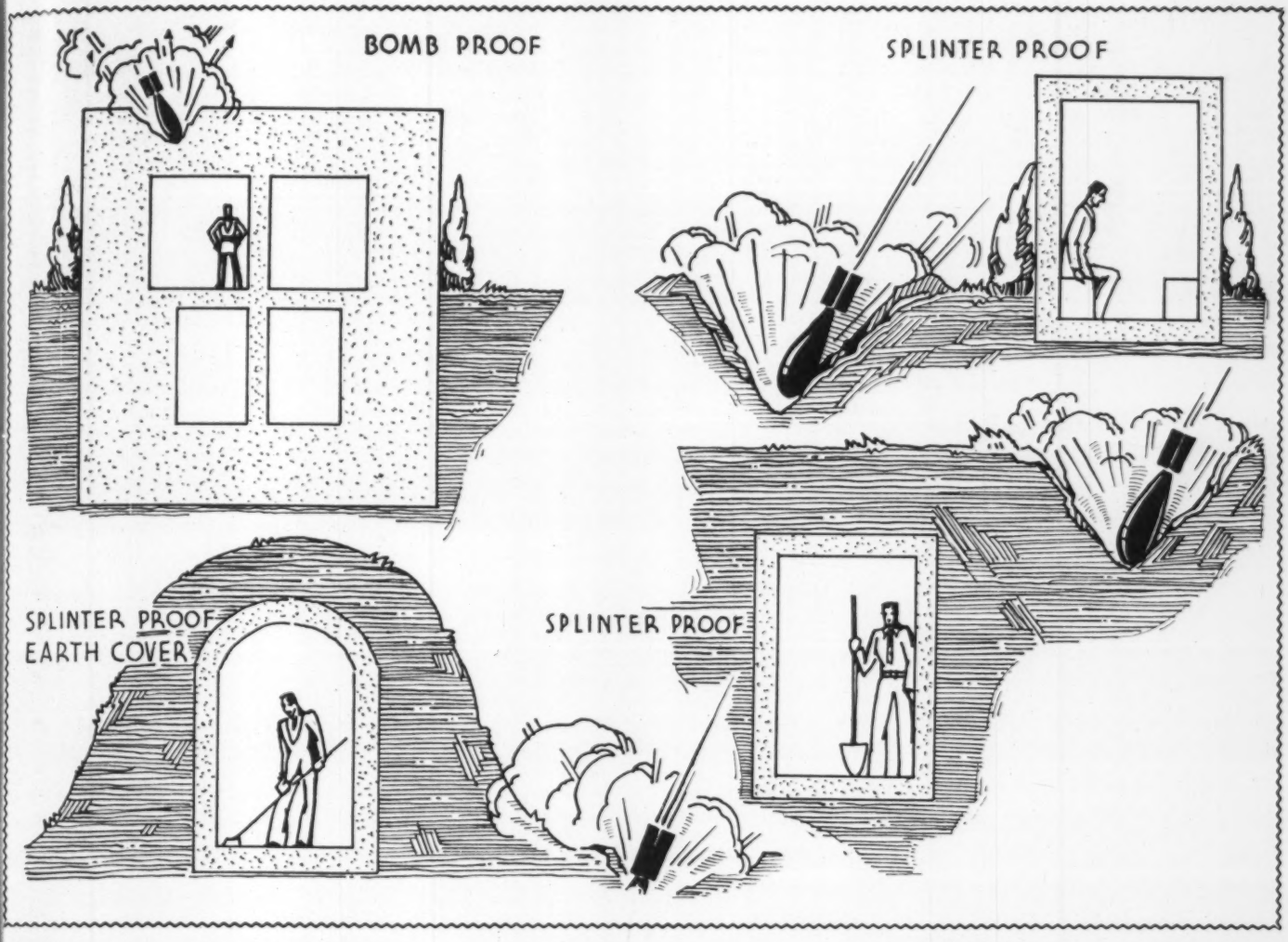


Fig. 1 (upper left)—Bombproof air raid shelter. . . . Fig. 2 (upper right)—Splinterproof shelter. . . . Fig. 3 (lower left)—Splinterproof shelter with earth cover. . . . Fig. 4 (lower right)—Buried shelter

★ HEAT and VENTILATION in

AIR RAID SHELTERS

• William J. McGuinness Shows Need of Mechanical Ventilation and Filters •

SUMMARY—In September, 1940, a course in aerial bombardment protection was incorporated into the curriculum of architecture at Pratt Institute, Brooklyn, N. Y., and is now a fundamental and permanent part of the training for the degree in architecture given at Pratt. The United States Office of Education in January, 1941, authorized the extension of this work to courses given for practicing architects . . . Mr. McGuinness manages these courses and lectures on the subjects of structural design and mechanical equipment of shelters. He confines this article to the heat and ventilation problems, and concludes that mechanical ventilation, with filtration, is advisable in all cases

PRELIMINARY TO a consideration of heat and ventilation in air raid shelters, it would seem important to decide on the general type and size of shelter in which problems of heat and ventilation will most frequently arise.

The bombproof shelter of Fig. 1 is too costly for general civilian use. Designed to stop a 500 lb medium case, high explosive bomb, it costs $500 per person sheltered, based upon actual records of shelters built in the United States. The next and probably the choice is the splinterproof shelter suitable for stopping the bomb fragments of a near miss and to resist blast and earth shock. Occupancy of this type of refuge is quite safe during the hit and detonation of the 500 lb bomb at a distance of 25 ft. The shelter can be built on the ground or completely underground. If placed over the ground, it will undoubtedly always have the earth cover shown in Fig. 3, which aids materially in stopping bomb fragments.

Contrary to common impression, there is no advantage to safety in burying the shelter. On the contrary, the effect of earth shock is more damaging than blast transmitted through the more elastic medium of air. The difference, however, is not

great. Both types are possible and they share quite evenly in advantages and disadvantages. The surface shelter has a slight advantage during attacks by toxic gases, most of which settle in low spots. It is also safer against flood by underground water or by surface storm water. The buried shelter is in a better position to lose its heat in warm weather by conduction and dispersion to the relatively cool earth.

The size of the shelter is fixed by the fact that it is inadvisable to house more than 50 people in the same room; this facilitates control, localizes casualties, and presents small, scattered targets. When in groups for large numbers of people, these 50 person shelters may be placed about 25 ft apart. For the sake of simplicity we will consider only shelters such as these, built specifically for the purpose, and omit the subject of adapted or strengthened portions of commercial buildings or residences where the mechanical equipment becomes very special in each case.

Space and Volume Requirements

Certainly our best research for design of shelter equipment has been the bombing of London, and the best design standards are those of the British Home Office resulting from these bombings. Some of these recommendations are shown. Fundamental differences, however, in the climate of the British Isles and that of the United States must definitely be taken into account in adjusting these standards.

Some basic considerations will be the same. It is evident, for instance, that there is a minimum floor area for physical comfort which cannot be reduced. This is generally taken at about 3.75 sq ft per person. Since a person requires an absolute minimum of 18 cfh of air to continue to live, shelters which do not have mechanical ventilation must afford a volume of about 50 cu ft per person to sustain life for a maximum of 3 hr if the shelter has to be sealed against a gas attack. This length of possible occupancy is likely to be shortened by the increasing temperature and humidity.

The British Home Office requires that all shelters for more than 12 people be capable of being sealed against gas in the event of a gas raid; otherwise, the shelter will be useless in a gas attack, and may be the scene of panic if it cannot be evacuated promptly upon the approach of gas. To wear a gas mask and remain in an unsealed shelter during a gas raid is not advisable, because of the possibility of the use of mustard gas and similar gases which attack the skin. The occupancy, while wearing masks, of a shelter mechanically ventilated by gas-laden air is equally poor policy. Panic may result.

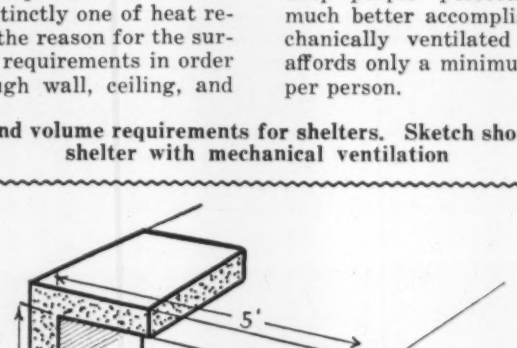
By studying the table in Fig. 5, it will be seen that in shelters not provided with mechanical equipment, it is necessary to provide greater floor area, greater volume, and greater wall surface. This carries two disadvantages. First, the cost of the larger shelter structure per person is much greater for naturally ventilated space: second, and far more important, by actual floor space the larger, naturally ventilated shelter will accommodate four times the number of people it has been designed for, shortening the possible three hours "sealed up" time to less than one hour. This will occur frequently because during severe raids it is impossible to refuse admittance while there is still floor space for all who apply. It becomes apparent that the best procedure is to design for minimum floor space per person, resulting in a small shelter, and then to provide it with mechanical ventilation. It is evident that this smaller shelter for the same number of people presents a smaller target.

Heat Requirements

A check calculation of the loss of body heat will quickly indicate that it is unnecessary in most cases to provide heat at all for a shelter designed for minimum space occupancy except for initial warming up in extremely cold weather. The problem under most conditions is distinctly one of heat removal. This is the reason for the surface-per-person requirements in order to provide enough wall, ceiling, and floor area to conduct the heat to the supposedly lower outside temperature. Here we must make a change from British standards. The average temperature in London over long periods during the summer may be below 70 F. Under these conditions heat *can* flow by conduction from the inside of a *surface* shelter to the cooler outside air (referring now to types shown in Figs. 2 and 3). In the United States average summer temperature will nearly always be above a comfortable inside temperature with the result that heat flows into the shelter, making conditions worse. It is therefore wise to plan to have the shelter below ground where the heat flow by conduction can be outward through the walls to a ground temperature in the vicinity of 50 F. Our choice is now Fig. 4, and Fig. 6 indicates the reason.

The amount of heat produced by humans varies with their state of rest or activity. This is also true of the amount of oxygen required, the carbon dioxide given off, and the amount of moisture by vaporization from the body. As between rest and violent activity, all of these things increase tremendously. The requirement to keep people perfectly still can be much better accomplished in the mechanically ventilated shelter, which affords only a minimum of floor space per person.

Fig. 5—Space and volume requirements for shelters. Sketch shows underground shelter with mechanical ventilation



MINIMUM REQUIREMENTS·PER·PERSON·

		FLOOR AREA 6 SQ·FT·	VOLUME 50 CU·FT·	SURFACE 75 SQ·FT·
OVER GROUND	NATURAL VENTILATION			
	MECHANICAL VENTILATION 150 Cu·F/Hr·Per Person	3·75	—	40
UNDER GROUND	NATURAL VENTILATION	6	50	75
	MECHANICAL VENTILATION 150 Cu·F/Hr·Per Person	3·75	—	20

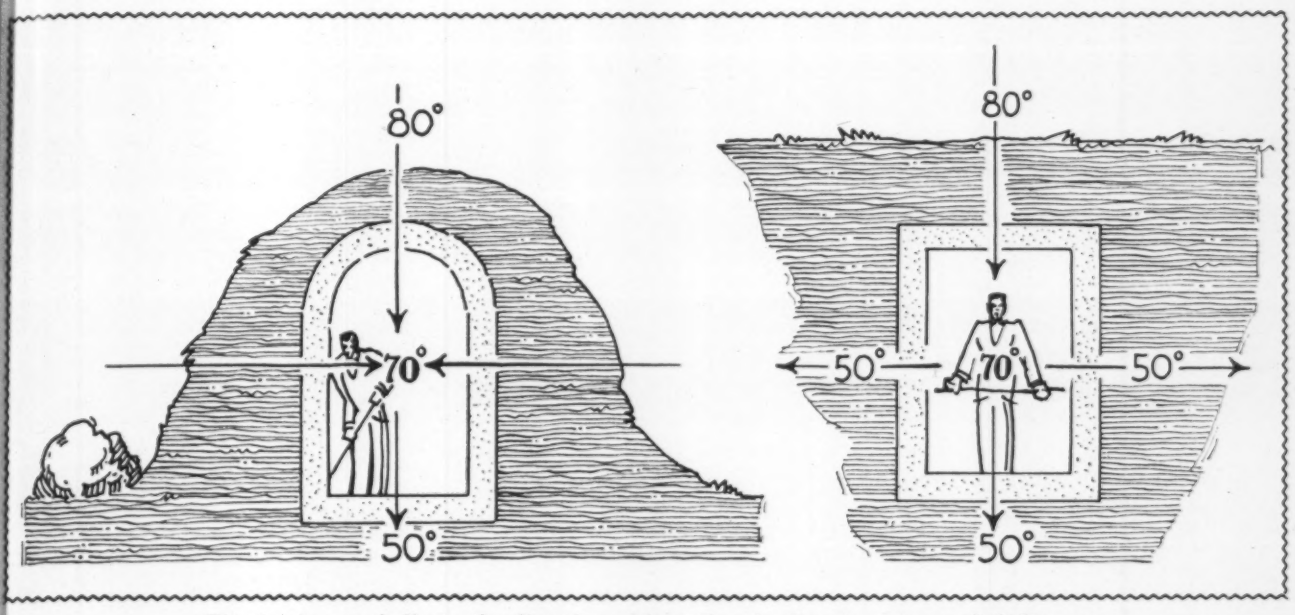


Fig. 6—Arrows indicate the direction of heat flow in these two types of shelters

Considering an underground *naturally* ventilated shelter without the advantage of positive ventilation or cooling, we can write a balance between the heat produced and the heat dissipated through the walls. Referring to the schedule in Fig. 5 we find a surface area of 75 sq ft per person called for. Assuming a desirable inside temperature of 70 F, a ground temperature of 50 F and a conduction coefficient of 0.70 for an 8 in. concrete wall, it is possible to lose 75x20x 0.70 or 1050 Btu per hr per person. At complete rest the person occupying this space produces an absolute minimum of 400 Btu per hr. Two things are evident. The occupant must remain perfectly quiet so as not to produce by activity three or four times this amount of heat, upsetting the balance; second, the thermal storage capacity of the walls and the adjoining earth will cause an increase in the 50 F temperature and a gradual lessening of the heat flow outwards. The inside temperature will certainly increase and without other aids there is a limited time during which the shelter may be occupied. This is usually taken as the three hours already mentioned. The same three hours is often used as a standard in mechanically ventilated shelters where the ventilation permits lower standards of floor space and surface. It would be expensive and probably unnecessary (because of the time limit of raids) to design for continuous use of the shelter. Twelve hours is occasionally taken as an upper limit with faster rates of circulation of air.

Under summer conditions in the United States, the circulating of summer air at 90 F will not help except perhaps in reducing the humidity and it would seem that cooling is mandatory in mechanically ventilated shelters. Actual installation of refrigeration is not common in England in spite of the elaborateness of some installations, but we may need it in America. Reasonably comfortable conditions *must* be maintained in shelters because with temperatures approaching 90 F and a relative humidity in the neighborhood of 90 per cent blood temperature will tend to rise, inducing fever and subsequent hysteria and panic.

At high temperatures, body heat can be lost by the evaporation of body moisture if the relative humidity is low enough. It is suggested that where the cost of cooling is excessive, careful consideration be given to the possibility of absorptive dehumidification of the air by calcium chloride.

If heat is needed before use to raise the temperature of the underground shelter from 50 to 70 F, it should be provided by some quick method such as an electric coil or the jacket water of a diesel used for power purposes in connection with the shelter group.

Shelter Ventilation

We have now tentatively settled upon a splinterproof shelter, placed underground, housing about 50 people, with 3.75 sq ft per person floor area, and having mechanical ventilation. A further advantage in mechanical ventilation will be seen by examining Fig. 7. With natural ventilation a difference in the height of the outside stack ends is necessary to induce a stack or chimney effect for air motion. In cool weather a sudden gas attack striking the low inlet grille might cause panic before the shelter could be sealed. With positive ventilation the air can be taken always from a height of 30 ft, where the concentrations of the heavy war gases are quite low.

Where there is any possibility of gas attack the mechanically ventilated shelter *must* be provided with gas filtration. Otherwise, as a sealed up shelter, it is four times worse than a naturally ventilated shelter having its proper occupancy under the same conditions. Assume then that the gas filter installation of Fig. 7 must be our system. The filtration unit consists of a fine mechanical filter for the removal of arsenical smokes and other toxic material in suspension. The air then passes through activated carbon as it would in a gas mask. The carbon absorbs the volatile war gases. The life of these units is short during attacks with heavy concentration and it will be necessary to have replacements on hand.

Because of the greater resistance of the filters it is wise to have a bypass for normal use during raids not using gas. If the air were continually passed through the filters they would become clogged with dust and extraneous matter in a short time and be useless when a gas raid came.

By keeping the fan near the intake it will be possible to minimize the length of duct within the shelter carrying gases which might leak dangerously into the shelter. Of course this length is normally under suction, but an opening in this duct would be a hazard as soon as the plant was stopped. The nature of the duct material will be different in shelters than ducts for normal use. We must think in terms of heavy gage sheet metal, fully welded at all seams. Also it becomes necessary to duplicate at separated positions the inlet and the exhaust openings to insure continuance of operation after one such position is bombed out. In the cramped conditions of a shelter, air motion is all-important and if there are dead spots it will be advisable to break them up with disturber fans.

Notice that there is a plenum created in the shelter which aids in excluding gases which might easily seep into the natural and regenerative

types of Fig. 7 even though sealed. It is thus much less necessary to gasproof the surfaces of a mechanically ventilated refuge. The exhaust can be through a toilet room or other similar space needing to have its air exhausted directly. At the top of the exhaust duct is placed a check type exhaust head, proof against the entry of gas and set for the correct plenum pressure.

The British recommendations for mechanically ventilated shelters are: For a three-hour occupancy, 150 cfh of air per person, and for 12 hr occupancy, 450 cfh of air per person. These will be recognized as quite low and for conditions in America will probably have to be increased and coupled with refrigeration or absorptive dehumidification or both.

Manually operated fans for breakdown service have been developed. One or two persons can easily take care of the ventilation and filtration for a 50 person shelter. The arrangement is one of crank operated or bicycle pedal operated equipment. Operators work on 15 min shifts. Diesels are very suitable for power use and the smallest unit will take care of several 50 person refuges. The diesel should be housed within a shelter but in a separate room. The quantities of combustion and cooling air are so great compared with those needed for the shelter, and the heat load so correspondingly large, that the air for diesel purposes must be separately introduced and exhausted. This air does not need to be gasfree.

The regenerative scheme shown in Fig. 7 is based on the use of oxygen tanks to revitalize the air, caustic potash to absorb the carbon dioxide, and calcium chloride to absorb the moisture. Because of the need for a skilled operator and the lack of an adequate positive pressure to exclude gas leakage, this type is not recommended.

These disadvantages of the regenerative system and of natural ventilation tend to indicate the advisability of mechanical ventilation, including filtration, in all cases.

"COME WORK IN AN AIR CONDITIONED OFFICE"

The want ads in the daily papers are always of interest, reflecting accurately, as they do, general business conditions.

All the advantages of available positions are listed these days; typical of many appearing last month was the following: "Wanted—Stenographer. Experienced; permanent; air conditioned small office."

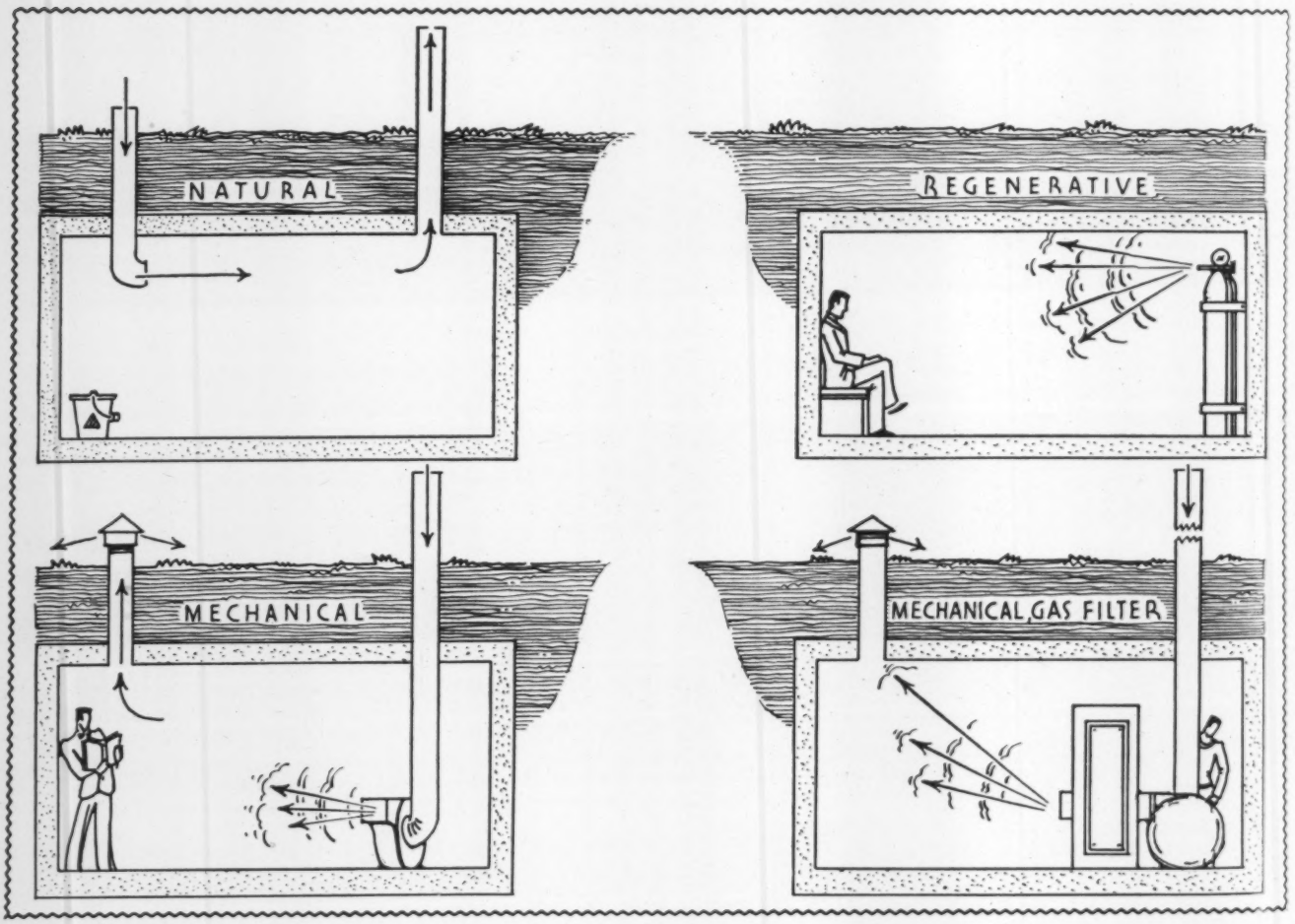
ORDER U-1 GOVERNS UTILITIES

The first basic order, No. U-1, of the Office of War Utilities, which was recently created within the War Production Board to administer war activities of electric, gas, water, steam, and communications utilities, replaces the former P-46 and all amendments and supplements thereto, re-enacting most of the P-46 provisions. It governs the flow of materials into the entire utilities field with the exception of communications, and is the first industry order to be integrated with the controlled materials plan.

An important group of provisions within the order, designed to effect sharing of material and equipment in utility inventories, requires utilities to sell surplus stocks as a condition for continuing to receive priorities assistance. These provisions are linked with the creation of regional surplus stock offices, through which utilities will carry on the redistribution of their surplus materials and equipment.

The order authorizes the use of a higher rating, AA-1, and a controlled materials plan allotment symbol, to permit utilities to obtain materials for maintenance, repair, and minor construction.

Fig. 7—Types of ventilation for air raid shelters



Heating-Piping-Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

Study of Actual vs. Predicted Cooling Load on an Air Conditioning System

By James N. Livermore,* Detroit, Mich.

IN THE design of an air conditioning system the amount of refrigeration to be installed and the sizes of the fans, ducts and cooling coils are based on certain calculations of the expected cooling and dehumidifying loads. The methods of making these calculations have been developed partly from theory and partly from laboratory tests without the benefit of much actual operating data. The author has long believed that this is an undesirable situation—that there should be more frequent checks made between operating results and design figures.

This paper reports some tests of the air conditioning system in an office building in which the refrigeration load was measured and separated by measurement or calculation, into its various components. The test results are compared with the design calculations and the discrepancies are analyzed and discussed.

The building on which this study is based is illustrated in Fig. 1. It was designed for the particular uses of certain departments of The Detroit Edison Company and is arranged to house offices, drafting rooms, electric meter shops, etc. Six floors are above grade and one below, the plan of each being a full rectangle approximately 120 ft by 250 ft. All corridors and utility spaces are located in interior bays. In some respects this is a loft type of building since it has many large open floor areas with a minimum of partitions. It also departs from the conventional office building in that its exterior walls are built of solid panels of glass block and masonry, leaving the structure virtually windowless.

Generally the interior artificial lighting is a coffer type in which lamps are set in open recesses in the ceilings as shown in Fig. 2. These coffers are arranged in a uniform pattern of nine units per bay and each is fitted with a lamp of capacity adequate to produce 50 foot-candle lighting intensity at desk level.

The entire building is both heated and cooled by three air conditioning systems[1] centered in the basement. The refrigerating plant illustrated in Fig. 3 consists of two motor driven centrifugal machines designed for chilled water service. One machine has 360 tons of refrigerating capacity, and the other, 180 tons. The output of each is controlled by manual changes in compressor motor speeds.

Hot water for building heating is provided by a heat exchanger to which steam is supplied from the district heating mains.

The air handling system is a double duct type and is arranged as shown diagrammatically in Fig. 4. Warm and cold air supply shafts extend in pairs the full height of the building in three central locations. Mixing dampers at each floor proportion warm and cold air from these shafts to supply the cooling or heating demands of thirteen or more control zones on each floor. At the base of each shaft is a supply fan. As the demand for cold or warm air varies, the static pressure is held constant on each shaft by controls which vary the position of fan inlet dampers. This is supplemented by manual changes of fan speed when required. From the central refrigerating plant chilled water is supplied to the cooling and dehumidifying coils at the three cold shaft fans and likewise from the steam heat exchanger hot water is supplied when required to the heating coils at the three warm shaft supply fans. Although the total quantity of air being supplied remains approximately constant at all times, it may be seen that the quantity of air through any one supply shaft, either cold or warm, will be continually varying with the demands of the mixing dampers it supplies.

Supply air leaving the mixing dampers is carried through individual zone ducts to diffusing areas in the room ceilings. A greater part of the building has perforated metal acoustic ceilings and portions of these are utilized as a means of introducing supply air to the occupied spaces. Air leaving the rooms travels to the main return shafts either through openings in the window sills and then through the ceiling space of the floor below or through corridors direct to the return shafts.

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[1]Unusual Air Conditioning Problems Met in Designing New Glass Block Building, by James N. Livermore. (Heating, Piping and Air Conditioning, April, 1938.)

Fig. 1—The Detroit Edison Company building housing offices, drafting rooms, electric meter shops, etc.

Fig. 2—Interior view showing lamps set in open recesses in ceiling

In addition to the mixing damper, each control zone is equipped with a remote bulb thermostat arranged with the sensitive element in the office space, but with the instrument proper out of reach of the occupants. Each of these thermostats, 82 in all, is connected to a readjusting control known as the *reset* which is capable of automatically changing the setting of each in response to changes of outdoor temperature.

Procedure

In order to obtain significant data on the performance of this air conditioning system during the cooling season, it was important to conduct tests during weather approximating the conditions assumed as a basis for its design. In an effort to do this much unsuitable data was gathered, since many test runs were started only to have the following weather turn too cool for heavy load operation. Generally, the testing was done by three four-man crews working in shifts independent of the operating force, and the tests were run continuously for 48- and 72-hr periods.

Specifically, the object of this study was to measure the total cooling load as accurately as possible and to simultaneously obtain data on its components. In addition to the total load, therefore, data were to be collected on the load due to the following:

1. Solar radiation and heat transmission through roof and walls.
2. Electric light and power.
3. Occupants.
4. Ventilating air.

Fig. 5 shows schematically the arrangement of the apparatus used in measuring the total refrigerating load. This provided data for calculating the rate of heat *removal* from the chilled water as it passed through the refrigerating machines. Due to the equipment layout this was a simpler method than taking data on the rate of heat *gain* at each of the three main fan room cooling coils.

Generally the apparatus consisted of a means of measuring the chilled water flow and the temperature drop which took place. Since it was never necessary to operate both machines simultaneously, one flow meter and one temperature reading on the return side of the plant was sufficient. A recording flow meter had originally been installed in the chilled water piping which was to meter the total flow through the two machines. The accuracy of this instrument was established by water leg test on the recorder, and conformity of the nozzle design and installation to well established standards. The meter was also checked against the main building city water meter and the two were found to be within 1.5 per cent in agreement. Having accurate flow data, it was necessary only to measure the temperature drop of the water through the refrigerating plant to arrive at the total rate of heat removal.

The accuracy of water temperature readings was very important since each degree Fahrenheit drop represented approximately 60 tons of refrigeration when the large machine was running and 30 tons when the small machine was running. It was felt that readings with error of no greater than 0.1 F were necessary. To obtain this degree of accuracy, thermocouples were used in conjunction with a precision type potentiometer. To avoid temperature lag the thermocouples were installed in contact with the water. For their correct location in the pipe, a preliminary traverse was made, taking several readings over one cross section. These indicated that a single reading at the center of the pipe would give a fair average temperature indication and that the effects of stratification in the pipe could be neglected. Since the readings of water temperature in and out of the refrigerating machines could not be made simultaneously, special care had to be taken to observe a true difference between them. Thus, each observation consisted of a number of readings taken alternately between supply and return for a period of 10 min at approximately 30 sec intervals. Typical data taken for one load computation are shown in Table 1.

Fig. 3—Motor driven centrifugal machines designed for chilled water service

Solar Heat Gain

In early test runs corresponding indoor-outdoor surface temperatures were measured with a view to approximating a rate of heat gain through the building walls. Proper interpretation of the data, however, became so complex that it was decided to repeat a portion of the tests during the next cooling season getting complete data on solar intensities.

The following summer, therefore, an Eppley pyrheliometer was used to record continuously the intensity of the sun radiation perpendicular to a horizontal plane at roof level. Having this information, hourly intensities normal to the walls of the building were then computed, and used with the new ASHVE heat gain data[2] collected in Pittsburgh during the same season. In order to make proper allowance for diffused radiant energy or solar sky reflection to the shady sides of the building, empirical values appearing in the same paper are used in this report for purposes of load analysis. These values were spot checked but no continuous readings were made during tests. In order to make a proper allowance for the effect of Venetian blinds, a survey was made on a bright sunny day, and a detailed hourly record kept of the areas of glass block shaded by the Venetian blinds in each bay. The data thus obtained were then assumed to be representative shading for any day when sun was a load factor.

[2]ASHVE Research Report No. 1147 — Heat Gain Through Glass Blocks by Solar Radiation and Transmittance, by F. C. Houghten, David Shore, H. T. Olson, and Burt Gunst. (ASHVE Transactions, Vol. 46, 1940.)

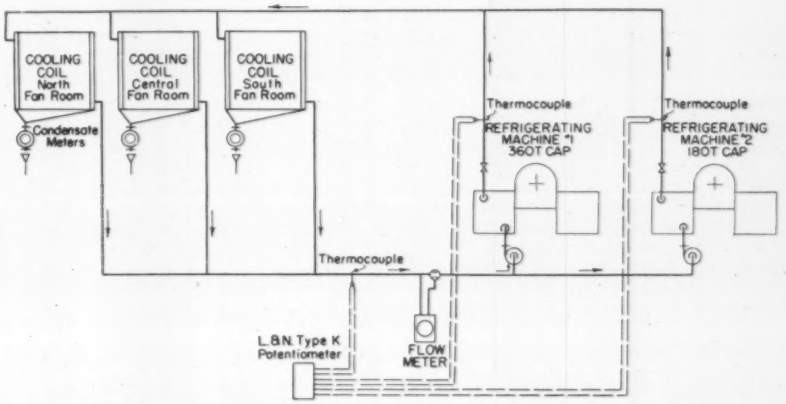


Fig. 5—Arrangement of test apparatus on chilled water circuit

Electric Light and Power

The interior load due to electric lighting and miscellaneous electrical devices throughout the building could not be measured directly. However, it was possible to read the total hourly electrical input to the building on the main kilowatt-hour meter. Also for the purposes of this test an electric meter was installed on every motor of 3 hp or over. The difference between the total building power and the total motor power then gave the kilowatt-hours consumed for lighting. The correctness of this method was checked by having a test crew actually observe the number and size of every lamp in use in the building each hour for several hours while another crew read the electric meters. This direct observation of energy consumed by lights agreed very close-

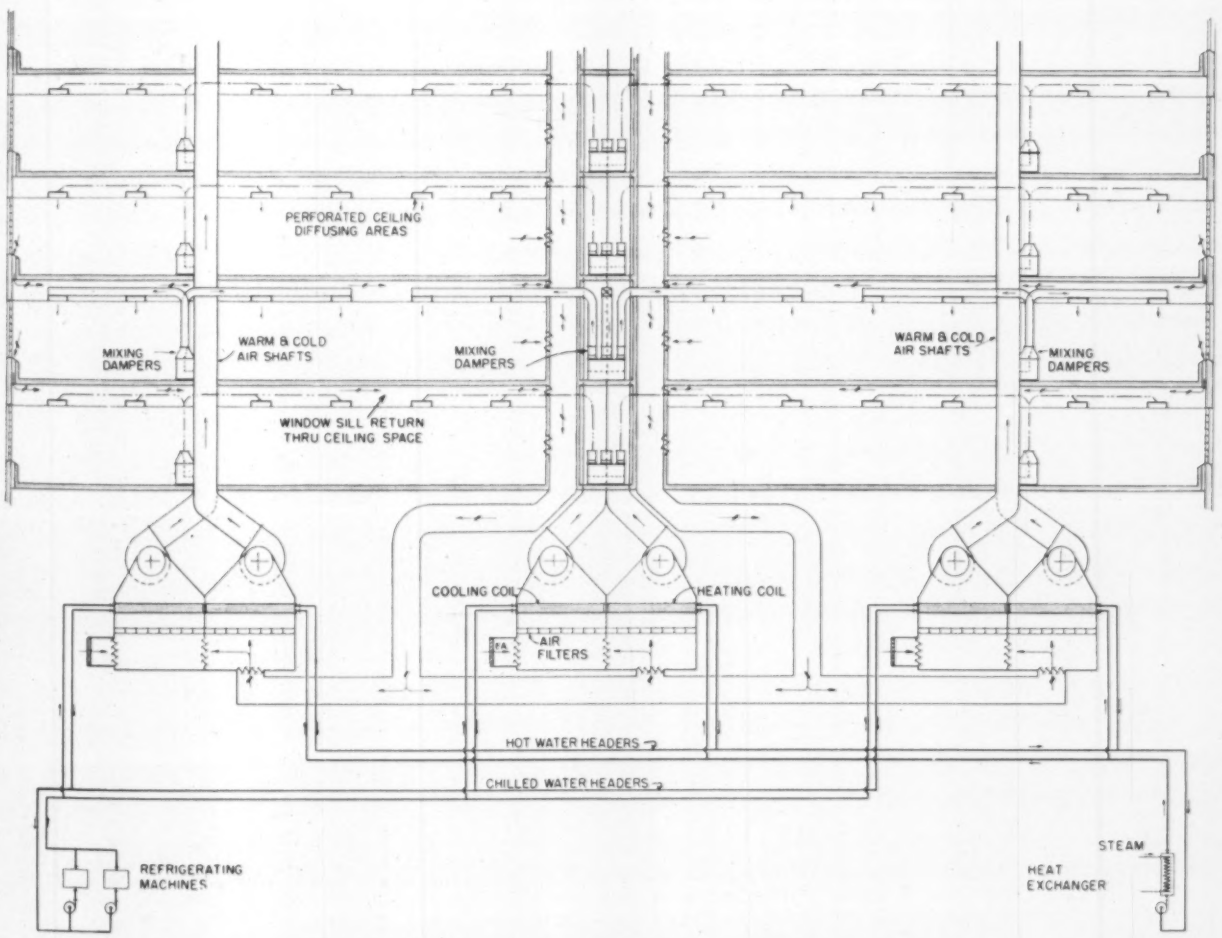


Fig. 4—Diagram of double duct type air handling system

Table 1—Typical Data Showing Temperature Readings for One Load Computation

Time	Chilled Water Temperature: Supply Temp. F	Return Temp. F	Temp. Drop F	Ave. Temp. Drop F	Chilled Water Flow: GPM
9:00 A. M.	43.8	49.2	5.4		810
	44.1	49.4	5.3		810
	44.3	49.7	5.4		810
	44.3	49.9	5.6		810
to	44.5	49.9	5.4	5.4 F	810
	44.6	50.1	5.5		810
	44.7	50.2	5.5		810
	44.8	50.2	5.4		810
9:10 A. M.	44.8	50.2	5.4		810

Table 2—Hourly Survey of Number of Occupants

Time	2 to 7 A.M.	8 to 9	9 to 10	10 to 11	11 to 12	12 to 1 P.M.	1 to 2	2 to 3	3 to 4	4 to 5	5 to 6	7 to 10	11 to 1 A.M.
Number of Occupants	20	701	671	615	460	477	630	636	617	334	38	10	25

ly with that measured by meter difference.

The majority of the motors referred to are located either in elevator penthouses or in the basement machine room, both of which are separated from the rest of the building and are not air conditioned. These data made it possible to exclude such motors in the heat source analysis unless they did contribute to the air conditioning load. The power consumed by chilled water pump and supply fan motors was itemized separately as a source of cooling load after a deduction had been made for motor losses.

Occupant Load

Since the number of occupants in the building varied from hour to hour through the day, but not from one day to another, a single survey of these was judged sufficiently accurate for the purposes of this test. Therefore, an actual count of occupants was made each hour throughout the building on a weekday. These data are shown in Table 2. Saturdays and Sundays the building occupancy was so low that it was neglected as a source of load.

Ventilating Air

The sensible and latent heat load due to outdoor air drawn into the fan room intakes is a function of rate of air intake and the difference between the dry- and wet-bulb temperatures of outdoor air and recirculated air. Data for these calculations were taken hourly. The air quantity had a tendency to vary and to facilitate measuring its rate of flow, baffles were installed in each intake to serve as metering orifices with manometers. Several rates of flow through these orifices were then measured with anemometers and the corresponding pressure drop through the orifices as indicated on the manometers was recorded. This gave data for a calibration curve for each manometer so that its readings could be used to determine the rate of flow readily.

Latent Heat Load

All moisture precipitated from the three cooling and dehumidifying coils was collected and carefully measured at the end of each hour of operation. With these data it was possible to calculate accurately the over-all latent heat load on the plant, this method completely avoiding the errors of instrument reading often in-

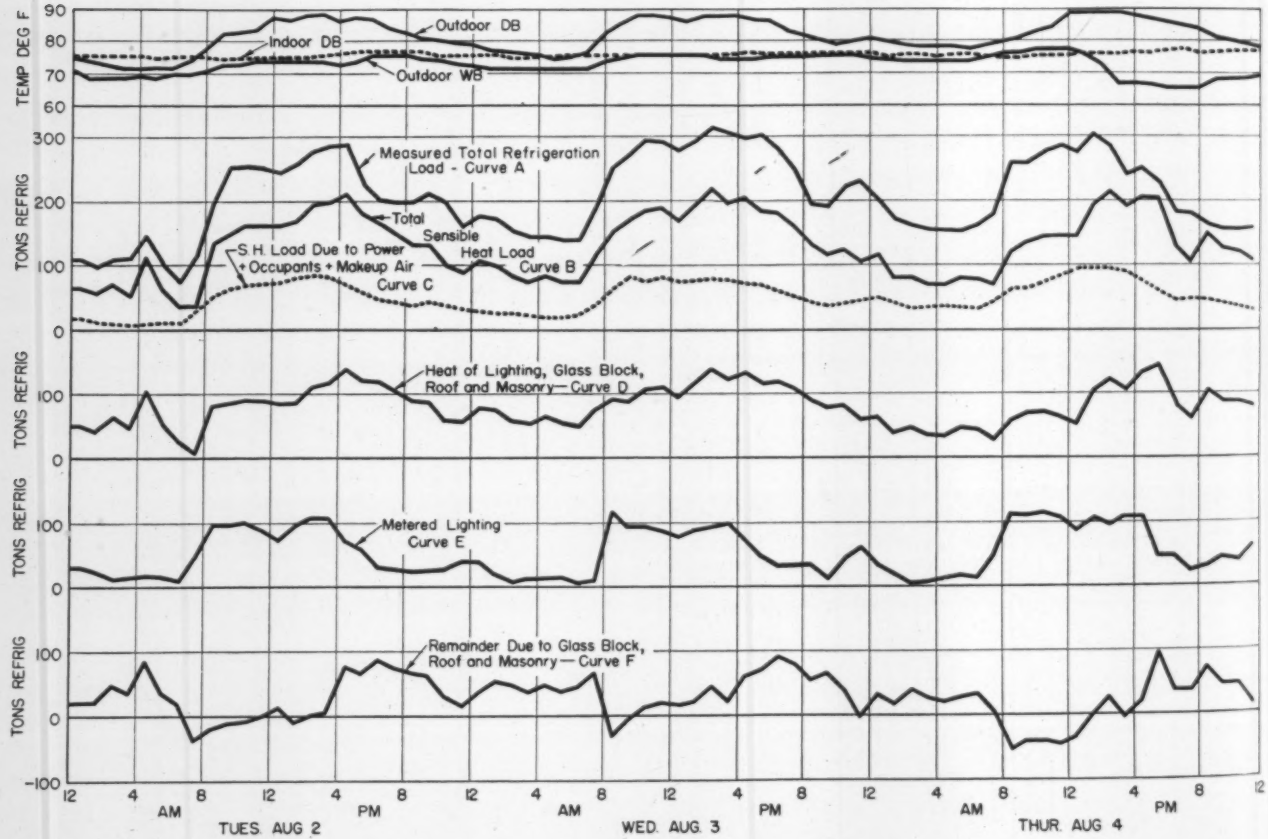


Fig. 6—Cooling load analysis (August 2, 3 and 4)

volved when wet-bulb temperatures are used.

Results

Analysis of Cooling Load

From the mass of data taken, the most typical of heavy load operation seems to have been secured during a 72-hr test made in early August. The uppermost curves of Fig. 6 show the temperature conditions during this run. It will be noted that at no time did the dry-bulb rise above 89 F, although on all three days the wet-bulb ran to 75 F and above, contributing greatly to the loads indicated. It is important to notice here that during this test the average building temperature was held fairly constant at 75 to 76 F from 8 a. m. to mid-afternoon, at which time it was allowed to rise. This practice has been adopted with a view to reducing temperature shock to employees on leaving the building at the end of the day.

Curve A is a plot of the measured total heat removed from the building by the refrigerating plant. The minor irregularities of the curve are largely due to manual changes in compressor speed and the consequent effect of allowing the temperature level of the chilled water to fluctuate slightly. The major trends of *Curve A*, however, are due to variation in load sources imposed upon the plant and an attempt to evaluate these is shown in the curves below.

Curve B is a plot of the total sensible heat load on the plant. These hourly values were accurately established by subtracting from the total cooling load, the latent heat of condensation of all the moisture collected from the cooling coil surfaces for each corresponding hour.

Curve C is a total of the heat due to fan and pumping power, plus the sensible heat given off by occupants, plus the sensible heat removed in bringing ventilating air from its outdoor dry-bulb temperature down to that of recirculated air. There are doubtless some inaccuracies in these values due to the method of counting occupants and a certain margin of error is to be allowed in measuring the air flow through the outdoor air intakes. However, the magnitude of these errors is believed to be less than 10 per cent of the totals represented by this curve.

If, from the total hourly sensible heat loads shown in *Curve B*, corresponding values in *Curve C* be subtracted, the remainder will leave heat values which can be attributed only to building illumination together with radiant and conducted heat passing through the building enclosure. An hourly plot of this remainder produces *Curve D*.

Curve E is an independent plot of the heat equivalent of the hourly electrical energy consumed in lighting the building. If these values now be subtracted from corresponding values in *Curve D* the remainder, shown in *Curve F*, can be attributed only to the heat gain through the glass block walls, the roof, and masonry around columns and floor spandrels, plus whatever errors may be involved in test measurements.

A brief inspection of corresponding indoor-outdoor temperature conditions will show that many of the heat flow quantities indicated in *Curve F* are quite obviously in error. On August 2 and 4 the values indicate (of course erroneously) that heat flowed *outward* through the walls from 7:00 a. m. until noon or after. While no data were taken on these days to permit close calculation of the rate of heat gain through the glass block, this material predominates in the wall areas, has a very short heat flow time lag, and the existing temperature differential and solar radiation would have made the indicated outward heat flow impossible. Other tests showed similar results when subjected to the same analysis with one notable exception: Fig. 7 shows a similar breakdown of the sensible heat load on a Saturday and Sunday when electrical energy used for lighting was at a minimum. On these two days no negative values were indicated for heat gain through the building enclosure even though the outdoor temperature was lower than on August 2, 3, and 4. Since the time when lighting was a minor heat source this sensible heat analysis gave more reasonable values for wall and roof load, there was a definite indication that the effect of lighting had been overestimated. While the heat equivalent of the kilowatt-hour cannot be challenged, the time required for the increase of one kilowatt-hour of heat released in a building to be felt by the cooling plant remains to be determined.

Sturrock[3] finds that approximately one-third of the total energy input to an incandescent lamp is the maximum that can be carried away directly by circulating air over it, and that the balance which is entirely radiant energy must be first intercepted by some heat absorbing surface before it can affect the air temperature. Since in this case the lamps are set in coffers and are not subject to any artificial circulation, it is probable that only an even smaller portion of this heat *immediately* affects the room air temperature. This is not to argue, of course, that the total energy input to each lamp does not ultimately become a part of the building cooling load (neglecting the small amount which may be radiated to the outdoors). It does seem probable, however, that the lamp heat effective at the refrigerating plant has a far different hourly load curve than that represented by the metered energy in *Curve E*.

In *Curve F*, Fig. 6, the greatest *negative* values for wall and roof load appear simultaneously with the beginning of heavy lighting load and diminish throughout the day until heavy lighting ceases. This gradual change seems to definitely indicate

[3]Effects of Artificial Lighting on Air Conditioning, by Walter Sturrock. (ASHVE Transactions, Vol. 44, 1938.)

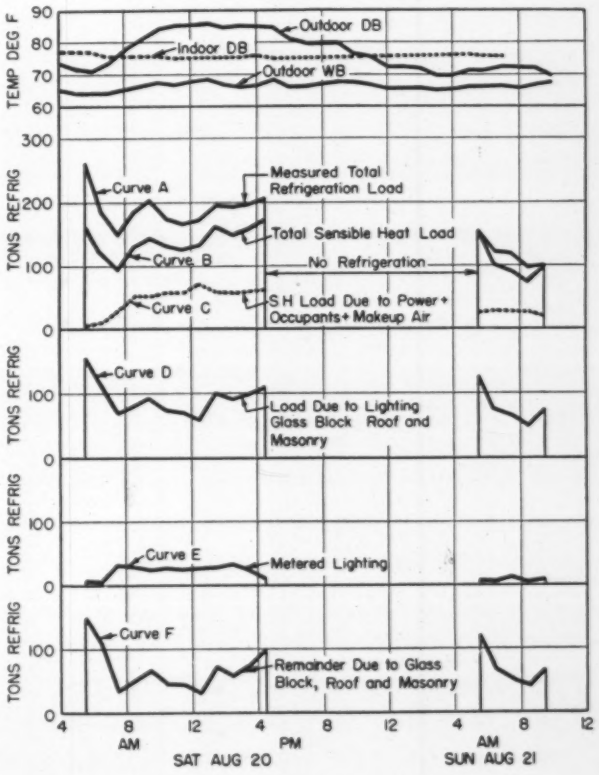


Fig. 7—Cooling load analysis (August 20-21)

that some lighting heat is being stored up and being released hours later. In view of these considerations it seems doubtful if the heat due to lighting can be treated simultaneously and at *full value* with other known heat sources, thus arriving at correct values for *Curve F*.

This failure to reconcile the total measured load with the sum of its parts seemed very important, particularly when all heat loads had been very carefully measured with the exception of that through roof and walls. Since this discrepancy might lead to certain changes in the calculation of cooling loads generally, a repetition of the test, obtaining at the same time data for accurately estimating the heat gain through the building enclosure, seemed well justified.

The analysis of a continuous 48-hr test, made through July 7 and 8 of the following year, is shown in Fig. 8. Although the outdoor dry- and wet-bulb temperatures were somewhat lower on the second day than on the first, the curves of solar intensities for the two days, which were virtually cloudless, are practically identical. Since July 8 was a Saturday, the building was practically vacant and a large difference between the two days is shown for load due to lighting and occupants.

Operating practice during this test had changed from that of the year before, particularly in the control of indoor temperatures. Where previously the setting of all room thermostats was varied automatically in response to changes of outdoor temperature, this method was abandoned in favor of resetting the thermostats manually from a remote central control point. Indoor temperature adjustments made by the operator had a striking effect on the load at the refrigerating plant as reflected in *Curve A*, Fig. 8. Here we see a continuous decrease in total load from 9:30 to 11:30 a. m. on July 7, even though the outdoor temperature rose from 86 to 90 F during the same interval. At the start of this period the building temperature was 75 F. From time to time during the morning hours the operator raised the thermostat settings, thus allowing the average temperature to rise approximately one degree. This permitted much of the interior heat to be absorbed by interior materials which were at a temperature lower than the new thermostat settings, and, in effect, unloaded the refrigerating plant. During the rest of this day (until 7:00 p. m.) the thermostat settings were left unchanged, and *Curve A* shows loads corresponding to constant temperature operation.

The procedure for analysis of the cooling load is much the same as that explained for Fig. 6. *Curve A* is the total cooling load for the building. *Curve B* is the total sensible heat load. *Curve D* is the remainder after the sensible heat load due to power, occupants and make-up air have been subtracted from *Curve B*. *Curve G* is heat gain through glass block, masonry and roof calculated on the basis of solar intensities observed during the test and with benefit of the latest information on the subject as it appears in the Heating, Ventilating, Air Conditioning Guide, 1942, pp. 141-149.

If the values in *Curve G* be subtracted from corresponding values in *Curve D* the remainder must represent the load due to the heat of lighting. Plots of this remainder appear in *Curve H*. These are to be compared to *Curve E* which represents the actual metered lighting load.

There are two explanations for the differences in the cooling load due to electric lighting indicated by these curves. The first is the likelihood of temporary storage of heat from incandescent lamps in the building material as previously suggested. Secondly, with the manipulation of the thermostats in the hands of the operator, it is possible for him to vary the load on the cooling plant without regard to weather conditions, *i.e.*, by raising the control settings, the plant is temporarily unloaded, and by lowering the settings the plant may be loaded beyond its capacity. This manipulation of controls plus cumulative errors in the data could easily account for the erratic relationship between the two curves. The significant part of the data then is that with all other heat sources carefully measured and computed, the heat due to lighting during the course of a day averages considerably less than the heat due to electrical input. In this case, on both July 7 and 8 the remainder due to lighting (*Curve H*) is 36 per cent less than the electrical input.

Curve F is of interest only to show again that if lighting load is taken at full instantaneous heat value, leaving wall and roof load as a remainder, the latter results in negative values just as in preceding tests, and the greater the lighting load the more extensive are these false remainders.

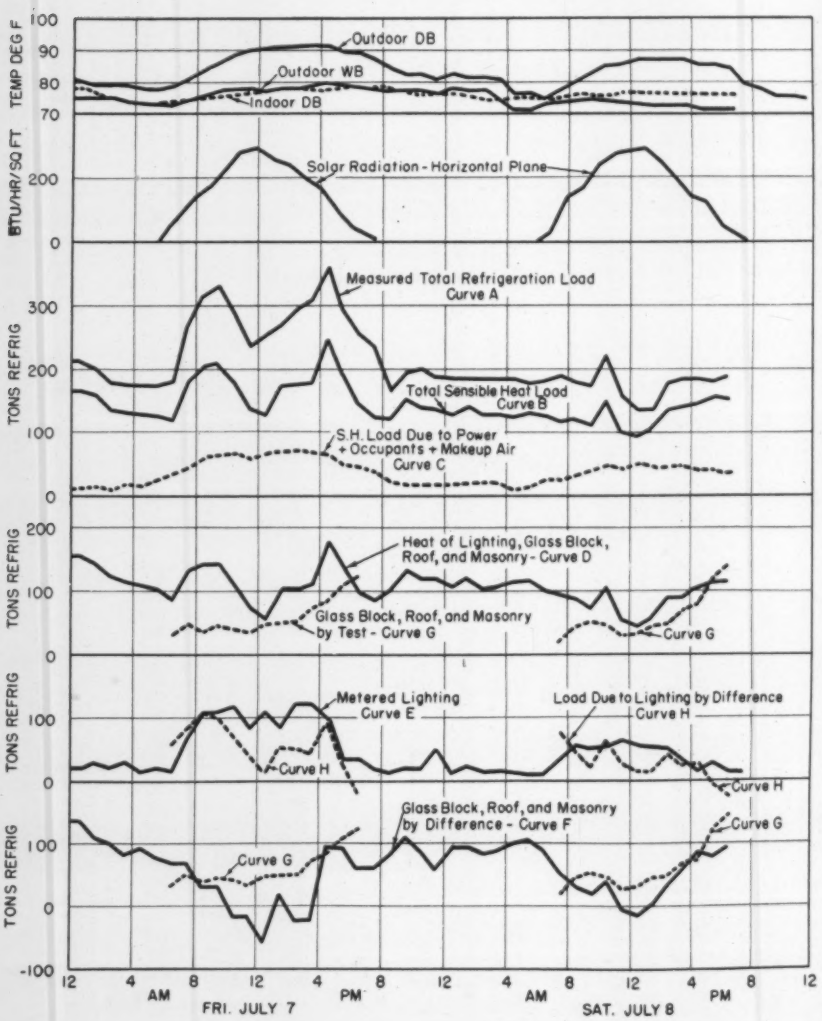


Fig. 8—Cooling load analysis (July 7-8)

Comparison Between Design and Test Conditions

None of the test work was done when weather conditions exactly duplicated those assumed as a basis of

design. However, a comparison between the operating conditions that were assumed at the time of design, and those which existed at the time of the test have a definite bearing on the findings of this investigation. It should be noted here that while the original calculations indicated that no less than 450 tons of refrigerating capacity would be required, at no time in the five years of operation has the cooling load been more than the 360-ton machine could carry.

In arriving at the capacity of the refrigerating plant, many assumptions were made as to the ultimate use of the building, and calculations were based on data which have since been considerably changed.

The following briefly compares the load components as they were originally calculated, with the same components using present data and corrected assumptions:

Glass Block: The original data on the solar heat transmission through glass block were given by the manufacturer as 54 per cent of an equal area of single glazed factory sash and calculations were made on that basis. New data on this material in the ASHVE Guide 1942 would result in the following differences between original and present calculations:

HEAT GAIN TONS

	9 A.M.	11 A.M.	1 P.M.	3 P.M.	5 P.M.
Original ...	78.0	52.4	79.0	105.5	79.0
Present ...	65.6	52.6	56.9	72.8	85.7
Surplus ...	12.4	—0.2	22.1	32.7	—6.7

Masonry and Roof: New data have been published on the time lag involved in transmitting heat through roof slabs and masonry. The original calculations were based on data[1] obtained by Sanford, Walker and Wells which showed a fairly constant rate of heat inflow for heavy masonry walls. Roof slabs were figured for a constant temperature gradient. The following difference would result from original and present data:

HEAT GAIN TONS

	9 A.M.	11 A.M.	1 P.M.	3 P.M.	5 P.M
Original ...	19.2	19.2	19.2	20.8	20.8
Present ...	8.6	8.5	21.7	29.9	33.9
Surplus ...	10.6	10.7	—2.5	—9.1	—13.1

Lighting: Original calculations assumed that 2700 watts of lighting would be installed in each bay, while in many locations this has been reduced to 1800 watts. It was also assumed that 40 per cent of the lighting would be continuously in use throughout a bright day. Comparison between this assumption and typical metered use is shown in last tabulation in left-hand column.

If, as the data in this test indicate, the *effective* heat equivalent of lighting averages to be 36 per cent less than the heat equivalent of electrical input, the comparison becomes:

HEAT GAIN TONS

	9 A.M.	11 A.M.	1 P.M.	3 P.M.	5 P.M.
Original ...	119.6	119.6	119.6	119.6	119.6
Actual	71.6	55.2	55.3	80.0	23.2
Surplus ...	48.0	64.4	64.3	39.6	96.4

Occupants: An ultimate building population of 1800 was allowed for. Although the entire building was in use at the time of the test, actual hourly count showed a maximum of 701 persons indoors at any one time and this number fluctuated widely. The comparison is as follows:

HEAT GAIN TONS

	9 A.M.	11 A.M.	1 P.M.	3 P.M.	5 P.M.
Original ...	60.8	60.8	60.8	60.8	60.8
Present ...	22.5	15.4	21.1	20.7	1.3
Surplus ...	38.3	45.4	39.7	40.1	59.5

The loads due to ventilating air and power for fans and pumping do not differ materially from those originally predicted.

These comparisons are summarized in Table 3, the loads in both cases being based on the following schedule of indoor-outdoor temperatures both for the original and present calculations:

Time	Outdoor Dry-Bulb	Outdoor Wet-Bulb	Indoor Dry-Bulb
9:00 A.M.	80.0	71.5	75.0
11:00 A.M.	86.1	73.5	78.2
1:00 P.M.	93.6	74.8	79.4
3:00 P.M.	95.0	75.0	79.4
5:00 P.M.	95.0	75.0	79.7

Indoor air dewpoint constant at 57 F.

Table 3—Comparison of Load Calculations—Original vs. Present

Time	9	11	1	3	5
Original Calculations			Load in Tons		
Glass block	78.0	52.4	79.0	105.5	79.0
Masonry & roof	19.2	19.2	19.2	20.8	20.8
Lighting	119.6	119.6	119.6	119.6	119.6
Occupants S.H.	34.0	34.0	34.0	34.0	34.0
L.H.	26.8	26.8	26.8	26.8	26.8
Ventilation S.H.	17.3	27.4	49.3	51.0	51.7
L.H.	72.2	79.5	67.2	66.2	60.5
Power	26.5	26.5	26.5	26.5	26.5
Total Load	393.6	385.4	421.6	450.4	418.9
Present Calculations					
Glass block	65.6	52.6	56.9	72.8	85.7
Masonry & roof	8.6	8.5	21.7	29.9	33.9
Lighting	112.0	86.3	86.5	125.0	36.2
Occupants S.H.	12.6	8.6	11.8	11.6	0.7
L.H.	9.9	6.8	9.3	9.1	0.6
Ventilation S.H.	17.3	27.4	49.3	51.0	51.7
L.H.	72.2	79.5	67.2	66.2	60.5
Power	20.8	17.6	23.3	22.4	24.0
Total Load	319.0	287.3	326.0	388.0	293.3

Effect of Fluctuations in Chilled Water Temperatures

At the time the system was designed, it was not known how close the control of temperature of the chilled water supplied to the cooling coils would need to be to assure satisfactory performance. Therefore, chilled water temperature controls had been specified as required refrigerating machine accessories. These devices depended for their operation on varying condenser water flow in response to slight changes in chilled water temperature. Such an arrangement is not economical of condenser water, which is important in this case since city water is used, but rather allows a wide variation in condenser water discharge temperature for any one compressor speed. An alternative, though not close, control of chilled water temperature can be accomplished with these machines by varying the compressor speed and manually adjusting the condenser water flow to some constant rate which will keep its discharge temperatures within a desired range. Better over-all operating economy could be obtained by this method and Fig. 9 shows to what extent its adoption affected the conditions in the building generally.

The rate of latent heat removal by the cooling coils varies with changes in any of several factors. Two of these factors, the dewpoint of the air entering the coils and the rate of chilled water supplied during this test were held fairly constant at the values noted. The changes in two other factors affecting coil performance are shown in the curves, namely, temperature of chilled water entering

HEAT GAIN TONS

	9 A.M.	11 A.M.	1 P.M.	3 P.M.	5 P.M.
Original ...	119.6	119.6	119.6	119.6	119.6
Actual use .	112.0	86.3	86.5	125.0	36.2
Surplus ...	7.6	33.3	33.1	—5.4	83.4

[1] "Field Studies of Office Building Cooling, by J. H. Walker, S. S. Sanford, and E. P. Wells. (ASHVE Transactions, Vol. 26, 1932.)

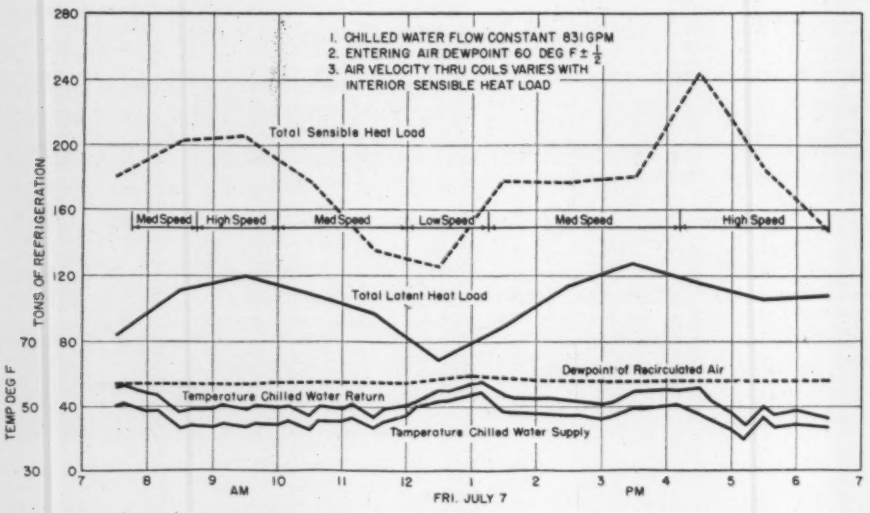


Fig. 9—Effect of variable chilled water temperature on dewpoint in occupied space

and leaving the coils and the rate of flow of air through them. With this system, the demand for cold air increases in response to a rise of room temperature above the thermostat settings; therefore the sensible heat removal curve indicates the variation in air flow through the coils. The intervals between changes of compressor speed are also shown.

The most important conclusion to be drawn from this comparison of data is that the chilled water temperatures do *not* have to be closely controlled in order to produce satisfactory room conditions. In this case, the water temperatures varied for short periods between levels as high as 54 F and as low as 40 F with little effect on the dewpoint of return air, *i.e.*, the average moisture content of the air within the building.

Effect of Shutdown

Generally it has been necessary on week days to maintain comfort conditions throughout the building until 1:00 a. m. to accommodate the night force, rather than shutting down at 5:00 p. m. when the offices are closed. These long hours of operation were not expected originally. For purposes of study, however, data were taken on loads encountered after an all-night shutdown lasting from 5:00 p. m. until 5:30 a. m. These data are shown graphically in Fig. 10. None of the shutdown periods shown extend through extremely warm weather and so do not show the longest time that might be required to attain comfort conditions when starting up again. Further, the curves of indoor dry-bulb temperature are for the recirculating air and, thus, represent the building average. While these show a maximum rise of 1½ F during a shutdown period, some zones on the top floor and on south exposures rose to temperatures as high as 85 F. More striking are the changes in wet-bulb temperatures and corresponding relative humidities of the return air which took place during shutdown. In one period the relative humidity rose to 63 per cent. This increase of moisture is not due to infiltration but rather due to continuous operation of the toilet exhaust system, which ultimately gets its air through the fresh air intakes. Since the building is practically air tight, this small amount of ventilation is desirable even though the building is not in use.

On both days shown in Fig. 10 satisfactory indoor comfort conditions were attained within a period of two hours after starting the air conditioning equipment. Other operating records show that in extremely hot weather, after a week-end shutdown, as much as eight hours may be required. It has also been found necessary to start the refrigerating plant during shutdown periods for dehumidifying purposes any time the indoor relative humidity rises above 50 per cent. This is done for the benefit of drafting and mapping departments where wide moisture changes seriously affect drawing and blueprinting materials.

No exact relation can be established between the extra ton-hours of refrigeration required for cooling down after shutdown, and the ton-hours that would be used in maintaining comfort conditions through the same shutdown period. Figs. 6 and 8, however, give some idea as to the nature of the night load when operating continuously night and day.

Conclusions

The following may be concluded from this test:

1. The importance of well considered design assumptions cannot be over stressed. Faulty assumptions may completely offset all improvements to data. Some of these assumptions, those that pertain to the *use* of the building, such as the number of occupants, are the responsibility of the owner of the building. Others are the responsibility of the designer. It is important also that *safety margins* in assumptions shall not be allowed to become cumulative thereby causing a gross over-estimate of load.

2. Due to the assumptions and approximations that have to be made before calculations can begin, extremely refined calculations for total cooling load are not justified.

3. In calculating the cooling load due to lighting, there should be a time lag allowance for heating up of building materials, with a varying proportion of the energy input going directly to the atmosphere. It is probable that where lights are surrounded by coffers as they were in this case, this time lag effect is accentuated. Further research on this point would be highly desirable.

4. Loads calculated for test conditions showed an erratic relation to those measured by test. However, the calculated peak loads were generally higher than the measured.

5. All loads seem to show a certain amount of lag or flywheel effect which is evidenced by the large amounts of heat to be removed at night when the plant is operated continuously.

6. Close control of temperature of chilled water to cooling coils is not necessary provided deviations from the required mean are not of long duration.

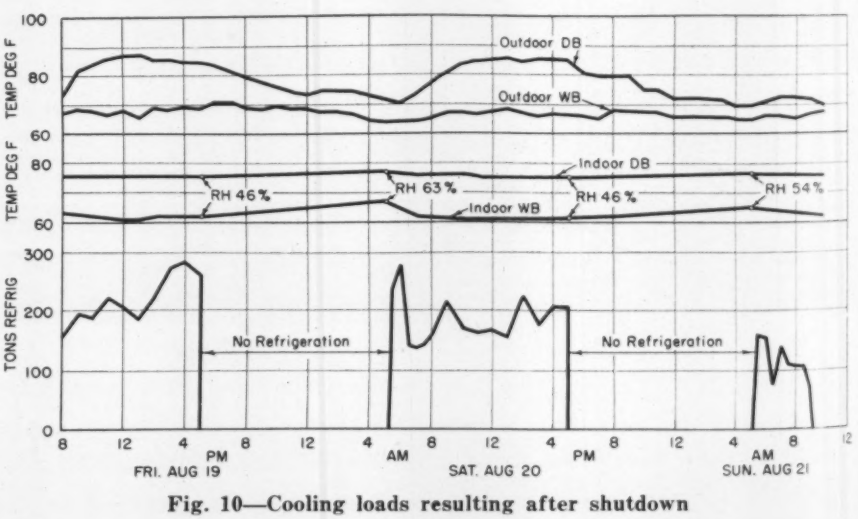
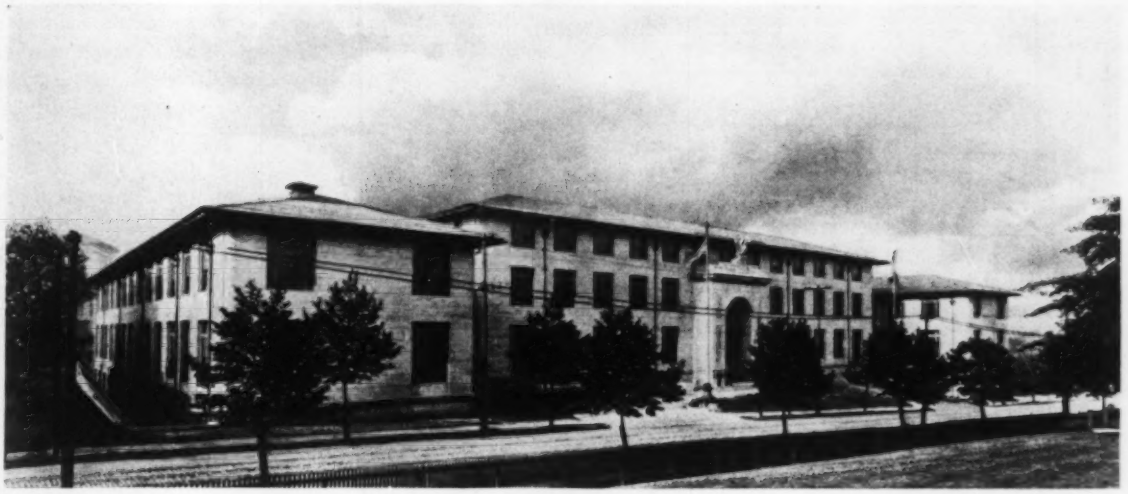


Fig. 10—Cooling loads resulting after shutdown



Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the American Society of Heating and Ventilating Engineers is located

Physiological Reactions Applicable to Workers in Hot Industries

By F. C. Houghten, Sc.D.,* Washington, D. C., Carl Gutberlet and M. B. Ferderber, M. D.,*** Pittsburgh, Pa.**

THE PAST decade has seen a growing interest in the study of industrial hygiene problems in many industries. Much of this interest has dealt with the atmospheric environment. In 1938, under the Technical Advisory Committee on Air Conditioning in Industry,† a research project was initiated by the Research Laboratory of the American Society of Heating and Ventilating Engineers to study the relation between the atmospheric environment of the industrial worker and his physiological reactions and degree of comfort.

A comprehensive report[1] of one phase of the study, including a program for continued work, was presented at the January 1939 Meeting of the Society. This paper described the summer season reactions of men engaged in light work to hot atmospheres with 60, 75 and 90 per cent relative humidity. A continued study, reported[2] to the January 1940 Meeting, gave seasonal variations in such reactions, indicating that acclimatization was a factor and that workers have the same reactions at lower effective temperatures during the cool seasons of the year.

Additions to the air conditioning systems serving the psychrometric chambers at the Pittsburgh Laboratory, permitting better control of low humidities, made it possible to extend the study to lower humidities and somewhat higher effective temperatures. The continued study aimed not only to give additional information on this phase of the subject, but also to check further the general assumptions that such reactions are functions of effective temperature.

Test Conditions

Generally, the same test procedure and the same type of subjects were used as in the earlier work. These arrangements and procedures are described at considerable length in the earlier report[1] and only the general outline will be repeated here.

The kind of work performed by the test subjects was designed to simulate many types of employment in modern industry where the worker's activity includes little more physical effort than watching an automatic machine and making slight adjustments or corrections, but where keenness of perception, sustained attention and readiness to act are important. To this end, a game of chance was introduced that required the close attention of the subjects while they stood or moved about, adjusting small weights as required in the operation of the machine.

Four college students, ranging in age from 18 to 23 years, served as subjects. They were selected carefully after physical and mental examination to determine their fitness for the work and whether their

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Presented at the 49th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, Ohio, January 1943.

[1]Superior numbers refer to appended bibliography of papers cited.

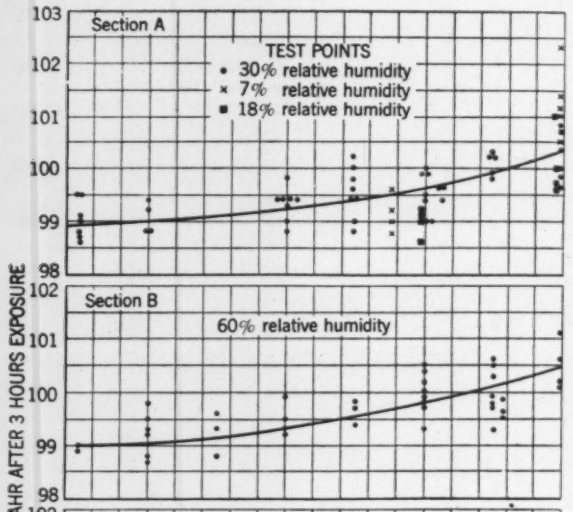

Fig. 1—Relation between body temperature and effective temperature of the environment for workers in 30, 60, 75 and 90 per cent relative humidities. A few tests at lower relative humidities are also shown

physiological reaction to hot atmospheres, including metabolism, was normal. Only those subjects whose basal metabolism varied little from a previously accepted normal of 40 large calories per square meter of body surface per hour were used.

To determine what constituted normal functioning for each subject, each was seated at rest for 45 min in the comfortable atmosphere of the control room before he entered the test chamber. During this control period and the following 3-hr test period (or a shorter period when the severity of the condition resulted in excessive physiological reaction) the subjects' body temperature, pulse rate, blood pressure, leucocyte count, vital capacity, presence or absence of perspiration, and feeling of warmth were determined at frequent intervals. Body temperature was determined rectally.

Tests were made over the effective temperature range from 77.5 to 95 deg, with a relative humidity of approximately 30 per cent. A single test was made at 95 deg ET with the lowest relative humidity that could be maintained by the air conditioning equipment (about 7 per cent) to serve as a check on the physiological reactions at the higher dry-bulb temperature and lower moisture content.

Test Data and Results

The body temperatures of each subject after three hours' exposure in the different conditions at 30 per cent relative humidity are plotted as points in section *A*, Fig. 1; a few points obtained at 18 and 7 per cent are indicated. For comparison, additional data collected in this study, and the results from earlier reports[1,2,3,4] for 60, 75, and 90 per cent relative humidity are plotted similarly in sections *B*, *C* and *D* of Fig. 1.

The body temperature for the different subjects in the various tests shows the usual spread which seems to be inherent in this type of physiological data. The body temperature, however, does show a consistent rise after 3-hr exposure in atmospheres above 85 deg ET, with a measurable rise for conditions above 90 deg ET.

The four curves from Fig. 1 are plotted for comparison in Fig. 2. There is a measurable spread of the curves for different relative humidities above about 90 deg ET, which would seem to indicate that a more pronounced temperature rise may be expected at higher humidities at the same effective temperature. A more careful analysis of the data in Fig. 1, however, seems to contradict this conclusion and emphasizes the probability that this spread is due to insufficient data and the chance falling of the points. In this connection, it is well to point out that the entire spread between the 60 and 90 per cent curves at 95 deg ET could be accounted for by a change of only a little more than one degree effective temperature. Similarly, a

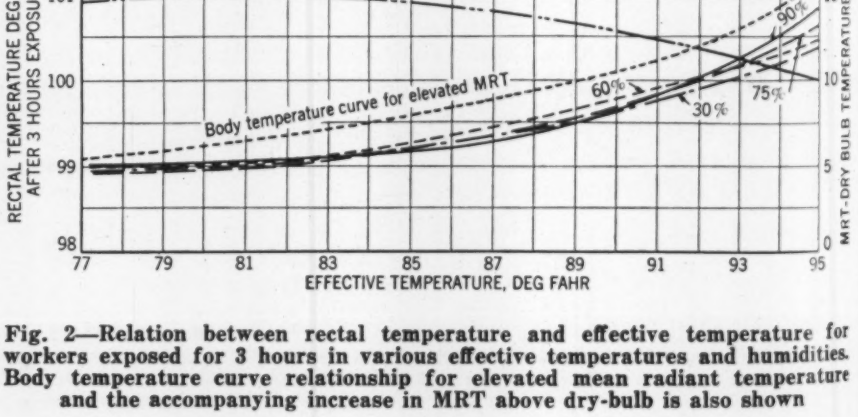

Fig. 2—Relation between rectal temperature and effective temperature for workers exposed for 3 hours in various effective temperatures and humidities. Body temperature curve relationship for elevated mean radiant temperature and the accompanying increase in MRT above dry-bulb is also shown

change of only 1.6 deg ET (or plus or minus 0.8 deg) would account for the entire spread between the 30 and 90 per cent curves at 95 deg ET.

For comparison, a curve plotted from tests at 60 per cent relative humidity and conditions having a mean radiant temperature higher than the dry-bulb temperature of the atmosphere is also shown in Fig. 2. The data from which this curve is drawn were collected under summer conditions and, therefore, the elevation in body temperature with the presence of radiant heat should be directly comparable with the summer data without radiant heat. The degree to which the mean radiant temperature exceeded the dry-bulb temperature of the air also is given. It should be emphasized that the data available on the effect of radiant heat are so limited that the results should not be considered conclusive; they do, however, add much to the meager present information on this important subject and serve to emphasize the importance of radiant heat in hot industrial environments and the need for a more conclusive study. It is of interest to note that the effect of the elevation of the mean radiant temperature on body temperature is about the same as the effect of one-third this number of degrees elevation in the effective temperature. In other words, about three degrees elevation in the mean radiant temperature seems to have the same effect as one degree elevation in the effective temperature.

All of the data reported in this paper and in the two earlier laboratory reports[1,2] are plotted in Fig. 3. Those for winter conditions[2] are indicated as circles and show the greater rise in body temperature in winter for the same environmental condition. For all of the variables affecting the data, including winter and summer conditions, variation

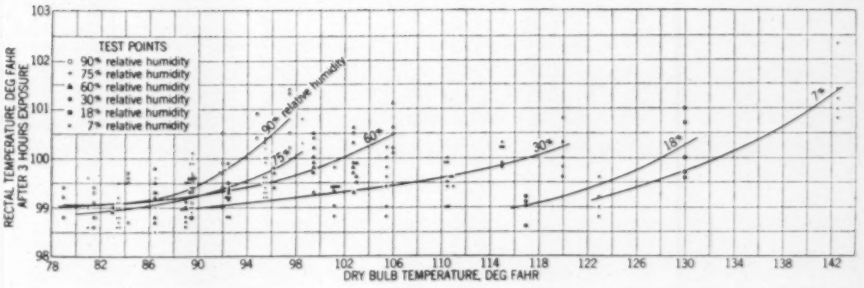

Fig. 4—Relation between body temperature and dry-bulb temperature for 7, 18, 30, 60, 75 and 90 per cent relative humidity (3 hour exposure)

in relative humidities, in individuals, and for the same individuals at different times, there is a spread of barely two degrees in the body temperature, or a plus or minus spread of less than one degree.

The relative correlation between the elevation in body temperature, and the dry-bulb temperature, the effective temperature index, and the wet-bulb temperature, may be seen by comparing the curves in Figs. 2, 4 and 5. The data plotted in Fig. 4 show conclusively that the dry-bulb temperature is of no value as an index to discomfort and physiological derangements in hot atmospheres. The data shown in Fig. 5 compared with those in Fig. 2 show that the effective temperature index is measurably better than the wet-bulb temperature index. This is particularly true when the data for 7 and 90 per cent relative humidities are compared in Fig. 5. Although the spread between the 30 and 90 per cent data is not great, it is consistent.

In the earlier study[1] of physiological reactions of workers to hot industrial conditions, the authors, as well as others considering the results, were surprised to find that the subjects engaged in light work while on their feet were comfortable and registered a given degree of moderate perspiration at higher temperatures than had been observed previously[3,5,6,7,8,9,10,11] for persons seated at rest in still air. This was explained by the authors on the assumption that the workers standing and moving about exposed more of their body surface area for heat dissipation to the atmosphere, and at the same time their movement produced effective air velocity.

The present study offered an excellent opportunity to confirm this fact. Both *at rest* subjects, clothed normally as persons usually are in an audience hall during the summer (*i.e.*, with shirt and lightweight summer coat), and *working* subjects used in this study (without shirt or coat), were exposed to the same atmosphere at 77.5 deg ET. The *at rest* subjects were seated at rest, and the *working* subjects performed their regular tasks. The *working* subjects under this condition were comfortable, while the *at rest* subjects with the greater amount of clothing and seated at rest registered comfortably warm, and warm.

A similar test was made under the same atmospheric conditions the following day, except that the *at rest* subjects did not wear a shirt or coat. In other words, both subjects were clothed alike. Again, the *working* subjects were comfortable, while three of the *at rest* subjects registered comfortably warm, and one, warm. The *at rest* subjects showed measurably more perspiration. This finding demonstrates clearly that the assumption offered to account for the apparent discrepancy in the earlier study[1] was correct; that is, a person may dissipate more easily a slightly greater amount of heat resulting from a higher metabolism when he is on his feet and working than when he is seated at rest.

Vital capacity, blood pressure measurements, and leucocyte counts were made as in earlier studies.[1,2,4]

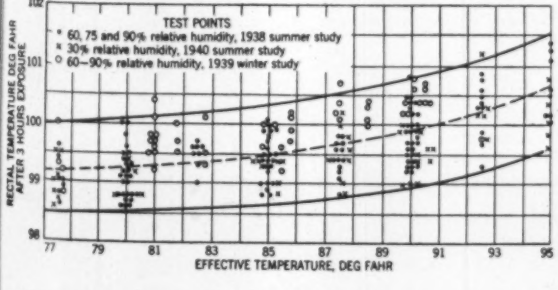
TEST POINTS
60, 75 and 90% relative humidity, 1938 summer study
30% relative humidity, 1940 summer study
60–90% relative humidity, 1939 winter study
RECTAL TEMPERATURE DEG FAHR AFTER 3 HOURS EXPOSURE
EFFECTIVE TEMPERATURE, DEG FAHR

Fig. 3—Relation between rectal temperature and effective temperature after 3 hours exposure. Points given for several studies

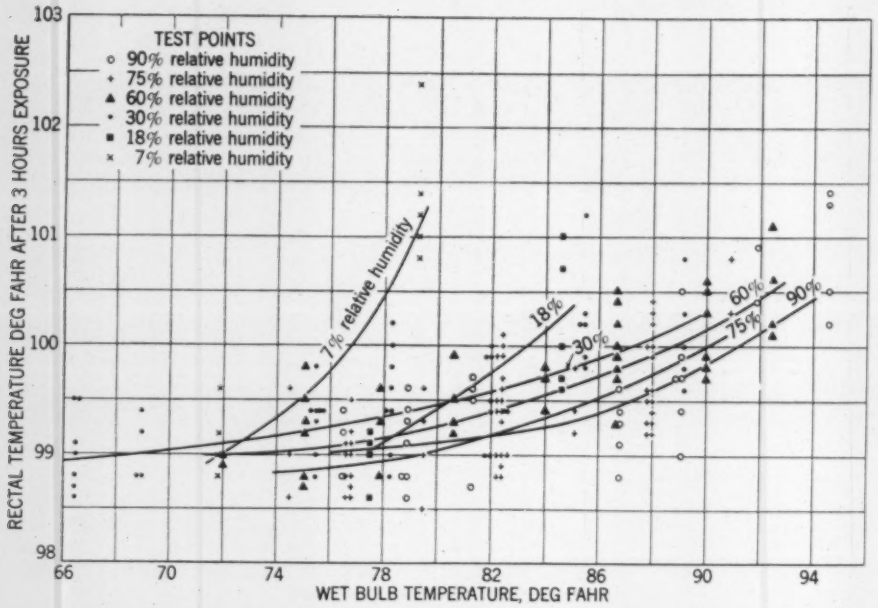

Fig. 5—Relation between body temperature and wet-bulb temperature for 7, 18 30, 60, 75 and 90 per cent relative humidity (3 hour exposure)

However, as these data are less consistent than those for temperature rise and a greater mass of them is required for statistical analysis to show significant variation, there is little advantage in including them here. The same rapid rise in leucocyte count, followed by a fall toward normal when body temperature returned to normal, was shown by the worker when exposed to hot conditions as in the earlier study.

Previous observations[4] revealed that frequent and continued exposures to hot atmospheres resulted in an apparent reduction in the response to automatic adjustment of the leucocyte count to a higher level with relation to a given hot condition. It is believed that this apparent fatigue of leucocyte control in a worker frequently exposed to hot conditions over an extended period has great significance.

Summary

1. A study of the rise in body temperature and increase in pulse rate of workers at relative humidities of 30 per cent and lower shows the same reaction at a given effective temperature as previously found for higher relative humidities.

2. The body temperature rise correlates well with effective temperatures, indicating a possible maximum, plus or minus, departure of less than one degree in effective temperature at 30 per cent relative humidity on a given effective temperature line in the neighborhood of 95 deg ET.

3. Rather conclusive evidence is reported to substantiate the earlier findings of the Laboratory, that a person is comfortable at a slightly higher effective temperature when standing and moving about at work than when seated at rest.

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2. Seasonal Variation in Reactions to Hot Atmospheres, by F. C. Houghten, A. A. Rosenberg and M. B. Ferderber, M.D. (ASHVE Transactions, 1940, Vol. 46.)

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9. Thermal Exchanges Between the Human Body and Its Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant. (The American Journal of Physiology, Vol. 88, April, 1929.)

10. Heat and Moisture Losses from Men at Work and Their Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant. (ASHVE Transactions, 1931, Vol. 37.)

11. Thermal Exchanges Between the Bodies of Men Working and the Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant. (The American Journal of Hygiene, Vol. XIII, March, 1931.)

Discussion

C. A. MILLS, M.D., Cincinnati, Ohio (written): Although Dr. Houghten and his co-workers, as well as others, had previously demonstrated that man does show adaptive changes when exposed to repeated or continuous difficulties in heat loss, they quite disregard this matter in their present study. No statement is made of the season or prevailing outdoor temperatures when the tests were conducted, nor do they indicate the chronological sequence of the tests made with each subject. Only 4 subjects were used throughout the numerous tests, so that the reactions of each one to the hot environment could well have suffered a considerable adaptive change before the conclusion of the study. Perhaps much of the spread in each group of readings would have been avoided if due attention had been given to this point. In any case, such reports of physiologic reactions should be carefully dated and the sequence of the tests indicated.

As I have often pointed out before, man is highly variable and adaptive in his reactions to changes in the ease or difficulty of body heat loss, and workers entering upon studies in this field should proceed with considerable caution.

W. L. FLEISHER, New York, N. Y.: This paper as presented by Dr. Houghten summarizes to some extent work that has been done by the Society over a period of years; and, it also gives further corroboration to the energy theory which was advanced by me and my son in a paper presented 2 years ago (ASHVE Transactions, Vol. 47, 1941, p. 175).

One thing that is lacking in this paper but is included in our data relates to the points which indicate the temperatures reached after a three-hour period. A careful analysis of the points which are presented shows that a great many are reached long before the end of the three-hour period, and remain stationary during the entire period, or for two-thirds of that period.

In other words, the reader may get a misconception from the curves of what actually occurs. The curves are meaningless, but the dots have meaning. One thing that is interesting is that the temperature of the subject becomes almost stationary at the same point. The rise in temperature, or the variations which Dr. Houghten indicated, are due to a great extent to the fact that, as he stated, people have differences in temperature in entering an enclosure. They may be 0.5 of a degree off. They may be as much

THE NEW GUIDE — 21st EDITION

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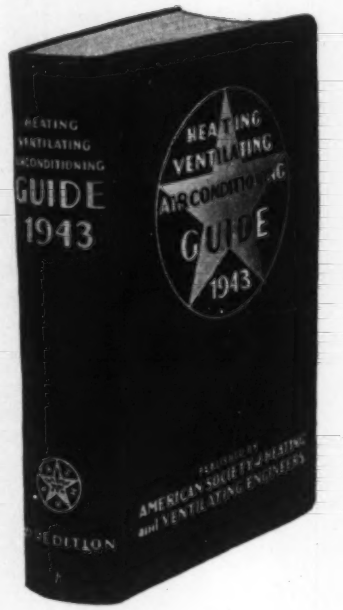


The Guide 1943

The 21st Edition—1943—of Heating, Ventilating, Air Conditioning Guide is off the press and available to men in the profession and the industry. In this volume is much new information valuable to men in the engineering branches of the armed services and men in industry who are concerned with problems involving heating, ventilating, and air conditioning.

This new edition contains 48 chapters of technical data and general information, including all phases of heating, cooling, ventilating and air conditioning, and related phases of refrigeration. Much entirely new data has been added and a large part of the book revised, rewritten and rearranged. Wherever possible technical data text matter has been condensed without detracting from a comprehensive presentation of the subjects.

Among the many subjects revised and brought up to date are: Thermodynamics of Air and Water Mixtures; Heat Transmission Coefficients; Heating Load; Estimating Fuel Consumption; Radiators and Convectors; Hot Water Heating Systems and Piping; Mechanical Warm Air Systems; new data on Unit Heaters, Ventilators, Humidifiers, Air Conditioners, Air Coolers and Attic Fans; and new data on Air Cleaning Devices where lint is an important factor. Other important revisions have been made to the chapters on Air Distribution, Air Duct Design, Sound Control, Motors and Motor Controls, Industrial Air Conditioning and Exhaust Systems, Radiant Heating and Design of Solar Water Heaters.

A new chapter, entitled Abbreviations, Symbols and Standards has been added; and much valuable information on state laws and codes relating to heating, ventilating and air conditioning forms an entirely new section of the book. And, in tune with the times, an Emergency War Practices section, outlining methods for conservation of materials, has been appended to the Technical Data Section.

All of this valuable data is fully indexed to permit convenient reference. On a single page, for quick subject reference, is a Section-and-Chapter outline of the volume; on each chapter title page is a brief summary of the chapter contents; and these references are amplified by a 14-page detailed index. These indices give easy reference to the many subjects presented.

This 1943 edition was compiled under the supervision of the Guide Publication Committee composed of Alfred J. Offner, *Chairman;* Paul D. Close, S. Konzo, C. S. Leopold and G. L. Tuve.

An important part of this newest Guide is the revised Catalog Data Section, in which much useful equipment data has been supplied by manufacturers. This section is grouped into sub-divisions for convenience in locating data. It is profusely illustrated and is supplemented by an Index to Modern Equipment which greatly facilitates its use. This section is arranged, indexed, and coordinated with the Technical Data Section so that it forms a valuable source of equipment data which supplements the technical data and facilitates its practical application to every-day problems.

In a total of 1268 pages this new volume includes 872 pages of technical data; 264 pages of equipment data; and the Roll of Membership of the Society. It is bound in a blue cover with gold stamping, and is now being distributed to the Society membership. Individual copies are available at $5.00 per copy. Thumb-indexed copies are obtainable at $5.50 each for those who desire this additional convenience.

NEW TECHNICAL SECRETARY FOR ASHVE

The appointment of Carl H. Flink as technical secretary of the American Society of Heating and Ventilating Engineers has been announced by Pres. M. F. Blankin. In this position Mr. Flink will be responsible for the coordination of technical committee work, will assist in the compilation of codes and standards and work closely with the Guide Publication Committee in the preparation of the annual reference book of the Society. He will be at the headquarters office of the Society, 51 Madison Avenue, New York.

Carl H. Flink

Mr. Flink was born in Ironwood, Mich., and is a graduate of the University of Michigan with a B.M.E. degree in 1915.

During his 27 years of business and professional experience, he has been engaged in research and development work as chief engineer of the Institute of Thermal Research of the American Radiator Co., director of research for the American Gas Products Corp., and mechanical engineer in charge of product development laboratory of the American Radiator and Standard Sanitary Corp. His work included the development of equipment and tests of boilers, radiators, furnaces, air conditioners, oil and gas burners, hot water heating and control equipment.

Mr. Flink joined the ASHVE in 1923 and the American Gas Association in 1927. He has had many committee assignments for both organizations and has participated in the development of codes and standards relating to coal, oil and gas burning equipment.

He is a licensed professional engineer in the State of New York, a member of Tau Beta Pi, and during the first world war served with the Meteorological Division of the U. S. Army Signal Corps.

(Continued from page 208)

as one degree off. Consequently, those who have lower temperatures show a greater rise in almost every instance.

The consequence is, that with 90 so-called effective temperature, those people who can stand the ambient conditions at all, who are in normal health take in practically a uniform temperature, not a variation of temperature; and that this uniform temperature, which is higher than the initial temperature at which they entered the enclosure, remains constant, and corresponds almost exactly with the increase in kinetic energy which is coming into them from the higher ambient temperature.

In other words, the rise, we will say, of 1 deg in their body temperature, when multiplied by the mass of the body, will give an energy level which corresponds to the energy level coming in to them. Consequently, the further carrying on of this work will indicate, I think, to a certain extent, that the theory that was advanced in the paper on Fleisher energy lines will be upheld.

CHAPTER DELEGATES MEET AT THE 49TH ANNUAL MEETING IN CINCINNATI

Chairman Collins reviewed the activities of the Chapter Relations Committee and said that speakers had been furnished for 26 Chapters, that a canvass had indicated that chapter dues varied from $2 to $6 and that Chapter dinners varied in cost from 75c to $2.00. Mr. Collins then invited Chapter representatives to speak on the subject of conducting meetings under war time conditions. Comments on this subject were offered by Earl Carrier of Boston, M. B. Shea, Detroit, C. H. Randolph, Milwaukee, L. E. Seeley, New Haven, Conn., C. H. Pesterfield, East Lansing, F. E. P. Klages, Greensboro, N. C., B. G. Peterson, Omaha, G. E. May, New Orleans, F. E. Triggs, Des Moines, Iowa, B. W. Farnes, Portland, Ore., and L. T. Mart, Kansas City.

As the subject of fuel oil rationing had been a matter of interest to many Chapters Mr. Collins invited Dr. Woods, Chairman of the War Service Committee, to comment on this work. Dr. Woods briefly reviewed the work of the War Service Committee which had held meetings in Washington, D. C., and had followed up the offer of the Society to allot its research facilities to Government agencies. The Office of the Petroleum Coordinator requested the Society to indicate ways in which the Research of the Society and the experience of its members might help in the preparation of material on methods of reducing fuel consumption. The Committee prepared its recommendations and at a conference in the Petroleum Coordinator's office it was pointed out that there is a point beyond which, if you cut fuel, you close up the house. We are quite willing to talk about the different things you can do if you will tell us just how much of a cut you are thinking of. If it is 25 per cent, or maybe as high as 30 per cent, you can make it. But when you go somewhere beyond that, well, you will just be out of luck. It is like the man who does not get any more than a certain amount of water; eventually, he will die.

We then discovered the difficulties that confronted governmental agencies in trying to decide what the cut would have to be. At one meeting the Office of the Petroleum Coordinator indicated to us 16 variables that entered into the problem, and they had to set a value on each one of these. The first one mentioned was, "how many States shall we ration—what area? What transportation facilities shall we have available?" The answer to question No. 1 was required before we could answer No. 2. Some of the others were, "What will be the demands of the Army and the Navy? What transport are they going to take out of the picture?" and so on.

Dr. Woods said that the Society sponsored a Fuel Conservation program which included War on Fuel Waste Week which was endorsed by a number of State Governors and the Committee sponsored a pamphlet on Ten Ways to Save Fuel, which was widely distributed. Dr. Woods suggested that all members read the November issue of Heating, Piping & Air Conditioning, which contains a statement about the work of the War Service Committee and cleared up the misunderstandings about the fuel rationing formula. Neither the War Service Committee nor the Society sponsored the formula used. It was developed by the staff members of OPA.

In the discussion that followed Mr. Farnes described the plan developed for the Pacific Northwest and told of the publicity given to the Oregon Chapters' work. Mr. Mather of Philadelphia, outlined the plan being used under the auspices of OCD and the Fuel Conservation Committee for Eastern Pennsylvania. Mr. Mittendorff told of the procedure used in Illinois, where the Chapter was active in assisting the Fuel Rationing authorities.

A comprehensive discussion of the proposed amendments to Art. B-VIII of the By-Laws of the Society relating to the method of choosing Society Officers was carried on until adjournment at 11:15 p. m.

CIVILIAN DEFENSE TECHNICAL ADVISORY COMMITTEE APPOINTED

The Director of the Second Civilian Defense Region has announced the organization of a Regional Technical Advisory Committee with 16 members, who will be called upon to solve the many technical problems that are arising in connection with civilian defense. The personnel of this Advisory Committee is as follows:

Name	Position in Civil Life	Representative for
E. P. Goodrich	Consulting Engineer.....Chairman, National Committee on Civilian Protection in Wartime	American Society of Civil Engineers
C. A. Fuller	Consulting Mechanical Engineer ..... Slocum & Fuller	American Society of Heating and Ventilating Engineers
F. E. Brooks	Asst. Vice President.....New York Telephone Co.	American Institute of Electrical Engineers
J. A. Willard	Consulting Engineer.....Bigelow, Kent, Willard & Co., Industrial Management	American Society of Mechanical Engineers
B. F. Tillson	Consulting Engineer.....	American Institute of Mining & Metallurgical Engineers
Prof. A. W. Hixson	Head of Dept. of Technical Engineering, Columbia University	American Institute of Chemical Engineers
S. E. Armstrong	Engineer.....Maintenance of Way-System, New York Central System	American Railway Engineering Association
H. R. Dowswell	Architect	American Institute of Architects
Dr. H. van Zile Hyde	Senior Surgeon, USPHS..(R) Regional Medical Officer	American Medical Association
Dr. S. G. Hibben	Director of Applied Lighting ..... Westinghouse Electric & Manufacturing Co.	Illuminating Engineering Society
C. A. Holmquist	Director, Division of Sanitation ..... New York State Department of Health	American Public Health Association
W. W. Brush	Editor, Water Works Engineering	American Water Works Association
R. E. White	Spencer, White & Prentis, Inc., Engineers and Contractors	General Contractors Association
Major J. H. Brewster	Sanitary Engineer..... USPHS (R)	Office of Civilian Defense
H. E. D'Andrade	Regional Senior Engineer, Protective Lighting....	Office of Civilian Defense
Francis Keally	Regional Senior Engineer, Protective Concealment	Office of Civilian Defense

INDEX TO 1942 ASTM STANDARDS

A 225 page publication entitled Index to ASTM Standards, including Tentative Standards, has just been published by the American Society for Testing Materials. It is an adjunct to the 1942 Book of Standards and enables any of some 1100 standards and specifications to be located readily.

The Index is also of value in determining whether ASTM has issued standard specifications, test methods or definitions covering a particular engineering material or subject.

Copies of the Index are furnished without charge upon written request to ASTM Headquarters, 260 S. Broad St., Philadelphia, Pa.

BUY WAR BONDS

NAVY PROMOTIONS ANNOUNCED

Two Council Members now on active duty with the U. S. Navy have been promoted and their many friends in the Society will be glad to know that Captain A. E. Stacey, Jr., and Commander T. H. Urdahl have received advancement from their former ranks of Commander and Lt. Commander.

NEW OFFICERS ELECTED

A. C. Buensod, President, Buensod-Stacey Air Conditioning, Inc., New York, has announced the election of M. M. Smith, as Vice-President and Treasurer; G. A. Dornheim as Vice-President; and J. A. Lockran, Assistant Treasurer.

A number of the company's key men have entered the service and the total number of employees now on active duty in the armed forces is 20.

The former Vice-President, A. E. Stacey, Jr., is now Captain, USNR, and the former Treasurer, W. J. McDonald, is now Lieutenant, USNR.

BOSTON OCD EVALUATES AIR RAID SHELTERS IN EXISTING BUILDINGS

A method of appraising the value of air raid shelters in existing buildings and a suggested form for making a survey for establishing a rating in regard to safety are given in a report entitled *Evaluation and Establishment of Air Raid Shelters in Existing Buildings*, which has just been issued by the Office of Civilian Defense, Boston, Mass.

NEW MEMBERSHIP COMMITTEE REPORTS

The membership January 1, 1942, was 3132 and January 1, 1943, it was 3044—a loss of nearly 100 members, but that does not tell all of the story. Already 200 members are in service or in foreign countries who, for one reason or another, do not pay dues so their membership is inactive, thereby making our effective membership on January 1, 1943, 2844. In order to build up our effective membership, this year we should secure at least 300 more members than we lose.

The support of our research work, of our publications, and the continuance of our valuable contribution to the war effort depends on maintaining our membership.

The president of another society in a recent talk made the statement that many men are "just waiting to be asked," and I want to add to the statement that you are not asking a favor of anyone when you ask him to join our Society. You are doing him a favor in offering him a part in the work our Society is doing and a part in the present benefits that every member receives in proportion to what he gives.

I ask every member just to bear it in mind and say to the man he meets, "Why don't you join?"

E. K. CAMPBELL, *Chairman*
Membership Committee

Of interest to heating and ventilating engineers will be the table for determining the capacity of the shelters on the basis of wall space, volume, and ventilation requirements, which is as follows:

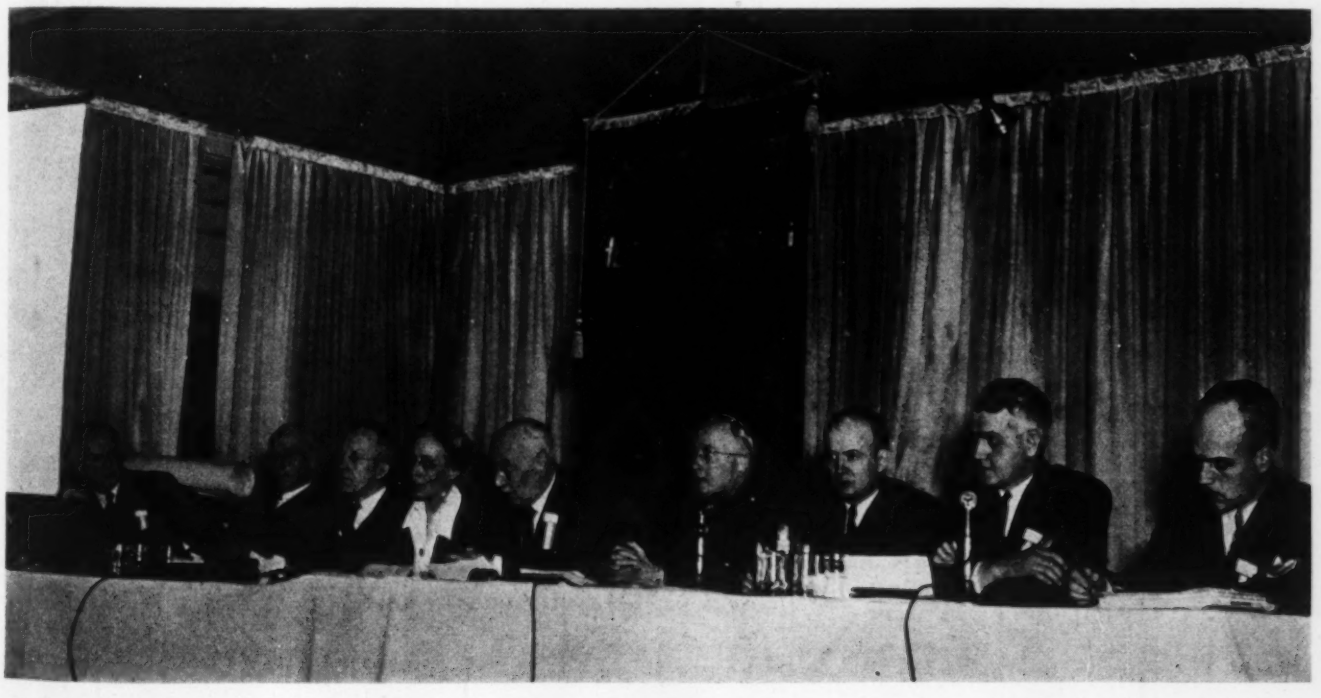
MINIMUM VENTILATION AND SPACE REQUIREMENTS IN SHELTERS

NATURAL VENTILATION *(open to outside air)*

LOCATION OF SHELTER	MAX. PERIOD OF OCCUPANCY (HOURS)	VENTILATION RATE (CU FT PER HOUR PER PERSON)	TOTAL SURFACE AREA (SQ FT PER PERSON)	FLOOR AREA (SQ FT PER PERSON)	VOLUME CONTENT (CU FT PER PERSON)
Above ground	3	.....	30	6	50
Above ground	12	.....	60	6	75
Below ground	3	.....	30	6	50
Below ground	12	.....	50	6	75
Above or below ground	3 12	Gas tight (none)	75 100	6	120 350
MECHANICAL VENTILATION[1]					
Above ground	..	300	40	8	60
Below ground	..	300	30	8	60

[1]Information furnished by the Chief of Chemical Warfare Service.

Note: The above figures are based on occupied space only; passageways, sanitary arrangements, entrance ways (gas locks, etc.), first aid and decontamination rooms are not included.



(Left to right) R. L. Davison, New York; Dr. C. A. Mills, E. K. Ruth, and Mrs. Ivah Deering, Cincinnati; Chairman C.-E. A. Winslow, New Haven; Lt.-Col. W. J. McConnell, Chicago; Allen Johnson, Primos, Pa.; R. A. Sherman, Columbus; and A. S. Bull, Minneapolis.

Members of Panel discussing the question of "How to Keep Fit in Cold Homes" at the 49th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cincinnati, January 26, 1943

SUMMARY OF LOCAL CHAPTER MEETINGS

Chapter	Meeting Date	Subject	Speakers	Other Features	Attendance	Attendance Ratio*
Illinois	March 8	Air Cooling, Ventilating and Heat Exchange Problems in Test Cells for Aircraft Engines.	H. E. Hardenbrook, Works Engineer, Buick Motor Div., Melrose Park, Ill.	Report of Tellers on Nominating Committee Election. Presentation of Pres. M. F. Blankin, Philadelphia; A. V. Hutchinson, Secretary of ASHVE; L. P. Saunders, Lockport, N. Y., member of Council and Chairman of Standards Committee.	109	0.80
	February 8	Heating the Ships for Victory.	John Clarke, Sr. Marine Engr., Head of Htg. and Vtg. Unit, Great Lakes Regional Office, U. S. Maritime Commission.	Certificate of Life Membership presented to Harry M. Hart and S. R. Lewis by O. J. Prentice, also a Life Member.	76	0.54
Kansas City	March 8	Strange and Unusual Patents.	C. Earl Hovey, Patent Lawyer, Kansas City, Mo.	M. M. Rivard announced that President Blankin would be the speaker at April Meeting.	26	0.43
	February 5	Business Trends, Economic Influences, and Probable Post War Effects.	Dr. T. Bruce Robb, Economist for Federal Reserve Bank of Kansas City.	Brief reports on 49th Annual Meeting in Cincinnati by L. T. Mart and W. A. Russell.	30	0.59
Northern Ohio	March 8	Cooling Towers.	L. T. Mart, Pres. Marley Co.		45	0.72
	February 8	Dehydration of Food Products.	H. W. Heisterkamp.	R. J. Meyer and F. H. Valentine guests of honor.	42	0.67
Pittsburgh	March 8	Coal.	H. P. Greenwald, Supt. Pittsburgh Experimental Station, Bureau of Mines.	Questions and Answers pertaining to the topic.	24	0.36
	February 15	The Sterilamp for the Reduction of Air-Borne Bacteria.	A. Paulus, Dist. Engr., Westinghouse Lamp Div.	Slides and Movies.	18	0.27
Ontario	March 1	A Panel Discussion on Pertinent Problems to Our Industry.	H. H. Angus. H. D. Henion. J. A. Clifton. R. A. Stencel.		57	0.57
Manitoba	February 24	General Business and Chapter Delegates' Report on 49th Annual Meeting in Cincinnati.	Ivan McDonald.		9	0.50
North Carolina	February 19	Military Explosives.	Dr. Marcus Hobbs.	Appointment of Nominating Committee to Report at May Meeting.	38	0.98
Iowa	February 16	A Little History on Pioneer Heating Men. Meeting in Cincinnati.	H. R. Holbrook, Iowa State College, Ames, Ia. F. E. Triggs, Pres. of Chapter.		10	0.32
Massachusetts	February 16	Comments on Recent Heating and Ventilating Installations in Industrial Plants.	E. L. Blair, Industrial Engr., Boston.		25	0.43

Chapter	Meeting Date	Subject	Speakers	Other Features	Attendance	Attendance Ratio*
Nebraska	February 16	Unique Heating Installation Adapted to Small Residence (combination of radiant and convection heating).	A. N. Ottum, Heating and Ventilating Engr.	W. A. Goll was appointed Chairman of Membership Committee to fill vacancy caused by N. M. Blanchard's Transfer to Milwaukee.		
		Proceedings of 49th Annual Meeting in Cincinnati.	B. G. Peterson.			
	January 25	Special Meeting. Review of article on Court House System Combines Panel Heating and Cooling and Conditioning.	E. F. Adams, Architectural Engr., Fort Crook, Nebr.	D. E. McCulley reviewed Chapter in ASHVE Guide on Radiant Heating. General Discussion on relation of humidity and temperature to comfort; apparent need for air movement; panel location; revisions necessary in methods of calculating heat losses.		
Michigan	February 15	Experiences Gathering Electrical Equipment for Edison Institute Museum.	J. W. Bishop, Ford Motor Co.	Music by Texas Ranch Boys—Ford Motor Co. Movies—A. E. Knibb.	81	0.85
Montreal	February 15	Food Dehydration.	C. C. Eidt, Dept. of Agriculture Central Experimental Farm, Ottawa.		34	0.55
Wisconsin	February 15	Direct-Fired Heaters in Defense Plants.	R. M. Rush.		36	0.50
		Report of 49th Annual Meeting in Cincinnati.	C. H. Randolph.			
North Texas	February 15	Post War Planning.	Glen Wasson, General Electric Supply Co.	Report from T. H. Anspacher as Delegate to 49th Annual Meeting in Cincinnati. Membership, Program and Entertainment Committees appointed. Board of Governors announced: T. H. Anspacher, L. S. Gilbert, R. K. Werner.	33	0.60
Philadelphia	February 11	Modern Practice in Hot Water Heating.	Merrill F. Blankin.	Life Membership Certificate presented to R. C. Morgan.	80	0.67
Southern California	February 10	Air Recovery Systems Used in Heating, Ventilating and Air Conditioning Systems.	J. S. Earhart.	Report on 49th Annual Meeting in Cincinnati by W. O. Stewart, delegate to meeting.	37	0.29
		Motion Picture Showing Foreign Bird Life in Aviaries.	A. G. O'Rear.			
Washington, D. C.	February 10	One Man's Opinion—an informal talk on application engineering as applied to heating, ventilating and air conditioning of the future.	Lieut. J. H. Van Alsburg.		20	0.20

Chapter	Meeting Date	Subject	Speakers	Other Features	Attendance	Attendance Ratio*
Cincinnati	February 9	Air Conditioning and Refrigeration in Aviation.	James Scott.	Report of Chapter Delegates Meeting in Cincinnati by A. Hard. Albert Buenger told of success of 49th Annual Meeting.	35	0.74
Western Michigan	February 8	Papers reported at the 49th Annual Meeting in Cincinnati.	C. H. Pesterfield, O. D. Marshall, V. H. Hill, T. D. Stafford, S. H. Downs.	Presentation of Life Membership certificate to W. W. Bradfield by S. H. Downs.	25	0.47
Golden Gate	February 3	Increase of Production Through Spreading of Work.	Capt. A. B. Court, U. S. N. (Retired) Inspector of Naval Material.	Meeting was a special occasion to honor men in service.	27	0.45
St. Louis	February 2	Operation and Servicing of Air Conditioning Equipment.	G. E. Becker, Frigid Refrigerator Service Corp.		23	0.40

*The attendance ratios shown represent the meeting attendance divided by the Chapter membership. These will be useful as a partial indication of interest shown by local chapter members in various types of subjects programmed by other local chapters, and may be helpful in deciding on subjects for other chapter meetings.

Candidates for Membership

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 14 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by April 15, 1943, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ASHCRAFT, J. PHILLIPS, Branch Mgr., Minneapolis-Honeywell Regulator Co., Dallas, Tex.	E. T. Gessell M. L. Brown	J. A. Ray R. G. Lyford
BERLET, E. JOHN, JR., Application Engr., York Ice Machinery Corp., Philadelphia, Pa.	C. S. Leopold M. F. Blankin	H. H. Erickson E. F. Roberts, Jr.
BONNER, JOHN, Heating Engr., Dayton Rubber Mfg. Co., Arlington, Mass.	R. S. Franklin W. T. Jones	James Holt C. T. Flint
BOWLER, ROBERT W., Student, Carnegie Institute of Technology, Pittsburgh, Pa.	C. M. Humphreys F. C. McIntosh	E. H. Riesmeyer, Jr. J. F. Collins, Jr.
CHIPMAN, EDWARD E., JR., Student, Carnegie Institute of Technology, Pittsburgh, Pa.	C. M. Humphreys F. C. McIntosh	E. H. Riesmeyer, Jr. J. F. Collins, Jr.
DALTON, ROBERT T., Sales Engr., State Supply Co., Cleveland, Ohio.	E. J. Sable C. H. Kaercher	E. W. Gray L. T. Avery
DINHAM, ROBERT E., Student, University of Minnesota, Minneapolis, Minn.	F. B. Rowley R. C. Jordan	C. E. Lund H. M. Betts
HEAVEN, LEWIS P., Owner, Heaven Engrg. Co., Kansas City, Mo.	H. H. Wright R. B. Mason	M. M. Rivard W. A. Russell
LILJA, OSCAR L., Htg. and Vtg. Engr., Northwest Airlines, Inc., St. Paul, Minn. *(Advancement)*	A. B. Newton C. E. Lund	William McNamara J. F. Stafford
MINOR, JAMES E., Student, Carnegie Institute of Technology, Pittsburgh, Pa.	C. M. Humphreys F. C. McIntosh	E. H. Riesmeyer, Jr. J. F. Collins, Jr.
RUDOLPH, ROBERT R., Naval Architect, Associated Shipbuilders, Seattle, Wash.	F. J. Pratt R. D. Morse	M. N. Musgrave E. H. Langdon

RUNNINGS, HENRY M., Mech. Engrg., Detroit Board of Education, Detroit, Mich.

SHER, ALVIN I., Student, Carnegie Institute of Technology, Pittsburgh, Pa.

WILHELM, JOSEPH E., Chief Engr., Avery Engineering Co., Cleveland, Ohio. *(Advancement)*

R. D. Randall
E. M. Harrigan

C. M. Humphreys
F. C. McIntosh

E. W. Gray
L. T. Avery

R. H. Oberschulte
H. E. Paetz

E. H. Riesmeyer, Jr.
J. F. Collins, Jr.

Philip Cohen
W. M. Rowe

Candidates Elected

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

ARVIDSON, ERNEST R., Electrical Draftsman, Carbide & Carbon Chemicals Corp., S. Charleston, W. Va.

BARNES, LEWIS L., Chief Engr., Carrier Atlanta Corp., Atlanta, Ga. *(Advancement)*

BOGGS, DENNIS, Engr., Avery Engineering Co., Cleveland, Ohio.

BULLER, CHARLES R., Chief Engr., The Heil Co., Milwaukee, Wis. *(Reinstatement and Advancement)*

CHURCH, LLOYD M., Dist. Mgr., Carrier Corp., Philadelphia, Pa.

COCKLEY, JONATHAN E., Jr., Electrical Engr., Public Service Company of Indiana, Inc., Greencastle, Ind.

COLVIN, OLIVER D., Comdr., Head of Shore Facility Br., Bureau of Ships, Navy Dept., Washington, D. C.

DONELSON, WILLIAM N., Engr., Modine Manufacturing Co., Racine, Wis. *(Advancement)*

DONNELLY, JOSEPH F., SR., Div. Mgr., Cleveland Steel Products Corp., Cleveland, Ohio.

EYNON, WALTER E., Pres. and Treas., A. C. Eynon Plumbing Co., Inc., Canton, Ohio.

GLANCE, ALVIN C., Field Engr., Anthracite Industries, Inc., New York, N. Y.

GRAVES, CLARENCE C., Engr., Robert Gordon, Inc., Chicago, Ill. *(Reinstatement)*

HOLLAND, WILLIAM T., Engr., Thomas E. Hoye Heating Co., Milwaukee, Wis. *(Advancement)*

KAPLUN, EUGENE, Designing Engr., Dwight D. Kimball, New York, N. Y.

KASAMEYER, ALFRED L., Engr., Detroit Edison Co., Detroit, Mich.

NOLAN, J. J., Owner, J. J. Nolan & Co., Memphis, Tenn.

NORBY, KARL H., Mgr., Htg. Dept., Tacoma Plumbing Supply Co., Tacoma, Wash. *(Advancement)*

POOLE, GORDON D., Dist. Engr., Can. Johns-Manville Co., Ltd., Montreal, Que.

PRIESTER, GAYLE B., Instructor Mech. Engrg., Case School of Applied Science, Cleveland, Ohio. *(Advancement)*

PURDUE, FRANK, Chief Engr., Odhams, Ltd., Watford, Hertfordshire, England.

ZINGHEIM, WILLIAM C., Owner, Wm. C. Zingheim Co., Chicago, Ill.

ASSOCIATES

BERRES, DANIEL S., Assoc. Naval Archt., Bureau of Ships, Washington, D. C.

BLAKE, JOHN L., Chief Engr., Air Cond. Dept., M.G.M. Studios, Los Angeles, Calif.

BRICKHAM, NELSON H., Assoc. Mech. Engr., Post Engineer, Ft. Warren, Wyo.

CORBIT, CHARLES A., Pres., Corbit's, Inc., Reading, Pa.

CORNELIUS, GEORGE E., Engr., Anthracite Industries Lab., Primos (Del. Co.), Pa.

CRANAGE, THOMAS, Sales Engr., Clarage Fan Co., Chicago, Ill.

CUPITT, ROBERT, Asst. Tech. Engr., Brockhouse Heater Co., West Bromwich, Staffs., England.

GOODMAN, ARTHUR L., Partner, County Appliance & Engineering Co., New York, N. Y.

HIMELBLAU, HARRY, Sec'y-Treas., Himelblau, Byfield & Co., Chicago, Ill.

LITTLE, KENNETH B., Mfrs. Repr., Kenneth B. Little Co., Cincinnati, Ohio. *(Reinstatement)*

MARSALIS, MARTIN E., Owner, American Metal Products Co., Fort Worth, Texas.

MARTIN, WILLIAM J., Pres. and Chief Engr., Economy Electric Mfg. Co., Inc., Cicero, Ill.

MAYER, WILLIAM J., Mgr., Engrg. Service Dept., A. M. Byers Co., Pittsburgh, Pa.

MCCALLUM, ALLAN W., Winnipeg Mgr., Anthes Foundry Ltd., Winnipeg, Man.

PROTHEROE, RICHARD A., Field Engr., Anthracite Industries, Inc., New York, N. Y.

SCOTT, DONALD C., Mech. Engrg. Consultant, Ludlow Mfg. & Sales Co., Ludlow, Mass.

TATE, HOWARD L., Chief Engr., Titche-Goettinger Co., Dallas, Texas.

THAYER, HARDING H., Htg. Engr., Blaw-Knox Co., Pittsburgh, Pa.

THOMAS, RICHARD L., Sales Engr., Fairbanks-Morse, Kansas City, Mo.

WEIR, FRANK F., Mgr., T. McAvity & Sons, Ltd., Winnipeg, Man.

JUNIORS

BASHAK, CIHAT M., Grad. Student, University of Michigan, Ann Arbor, Mich.

STUDENTS

BAKKE, GEORGE V., Student, University of Minnesota, Minneapolis, Minn.

FEILZER, JOSEPH H., Jr., Student, University of Minnesota, Minneapolis, Minn.

NORDSTROM, FIELDER A., Student, University of Minnesota, Minneapolis, Minn.

SNYDER, EDWIN J., Carnegie Institute of Technology, Pittsburgh, Pa.